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COMPARATIVE PERFORMANCE OF YEAR-AROUND SYSTEMS USED IN AIR CONDITIONING RESEARCH RESIDENCE NO. 2

by

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UNIVERSITY OF ILLINOIS BULLETIN

A REPORT OF AN INVESTIGATION

conducted by

THE ENGINEERING EXPERIMENT STATION
UNIVERSITY OF ILLINOIS

in cooperation with

THE NATIONAL WARM AIR HEATING
AND AIR CONDITIONING ASSOCIATION

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ABSTRACT

This bulletin is a complete report of results of year-around air conditioning investigations in Research Residence No. 2, a well insulated one-story frame house with a full-size basement. This investigation was conducted to study the effect of type and location of supply outlets and to determine operating procedure for obtaining best comfort conditions with each outlet type. Three supply outlet types were investigated; perimeter floor diffusers, ceiling diffusers, and floor registers located near inside walls. The performances of these three systems for both heating and cooling are compared with the performance of a fourth system investigated previously with which high sidewall registers were utilized.

The following are general conclusions which were drawn from the results of the investigation.

1. Perimeter floor diffusers were by far the best heating outlets. Room air temperature variations were small regardless of the supply velocity. Floor registers were the next best heating outlet followed by high sidewall registers and ceiling diffusers. At 70° F. indoor-outdoor temperature difference the three latter systems produced living zone (4-inch to 60-inch level) temperature variations in excess of the maximum of 5° F. recommended in a proposed ASHAE standard. Perimeter floor diffusers also produced the warmest floors, followed by the other outlets in the same order as listed above.

2. The total energy input to the residence, which included that supplied to the blower motor, household appliances, and lighting as well as fuel input to the furnace, was not appreciably affected by the air distribution system employed or the location of the supply outlets.

3. Ceiling diffusers were by far the best cooling outlets. Room air temperature variations were almost negligible regardless of the supply velocity. High sidewall registers produced the next smaller variations but cool draft areas resulted from the cool supply air dropping into the living zone. Floor registers with vertical non-spreading air jets produced comfortable conditions provided that the supply velocity was at least 500 f.p.m. Lower velocities were insufficient to project the cool air to the ceiling level and cause it to spread over the room. Instead, the cool air dropped to the floor and created cool areas. Floor diffusers were the most dependent upon supply air velocity of all outlet types. The characteristics of floor diffusers and floor registers were similar. The minimum recommended supply velocity for floor diffusers was also 500 f.p.m. but 600- to 700-f.p.m. supply velocity resulted in better room air distribution.

4. For year-around operation the correct choice of a system depends on the principal application. If heating is of major or equal importance, perimeter diffusers (the floor diffuser is but one type of perimeter diffuser) should be selected. The system should be designed for the optimum supply velocity during cooling. If cooling is to be the primary application and heating is of secondary importance because of relatively mild outdoor design conditions, ceiling diffusers will perform most satisfactorily. If a residence with either floor or high sidewall registers is being modernized, consideration should be given to utilization of the installed outlets.

5. The single, centrally located return air grille was sufficient and no bad effects were observed. Installation of a single return can largely reduce the duct installation cost. In larger homes it may be desirable to install more than one return air grille but it is not necessary to install one in each room since return air grilles have an insignificant effect on the comfort conditions. Provision should be made for the return air to move from the rooms to the location of the return air grille. This can be accomplished by undercutting doors or installing grilles in doors or walls.

6. Because of the unequal heating and cooling requirements of different areas of a residence some adjustment of the air-flow rate is usually necessary when changing from heating to cooling operation or vice versa. The most convenient method is to group ducts so that they may be closed by a damper to prevent overcooling or overheating.

7. Unless an unusually inexpensive source of cooling water is available the operation of waste-water-cooled condensing units is more expensive than the operation of air-cooled condensing units.

8. Night-air cooling is an effective means of reducing cooling unit energy requirements with mean daily temperatures below 78° F. The daily reduction in operation time was approximately 4 hours.

9. Forced attic ventilation is a less effective method of reducing the cooling load than addition of insulation. The maximum attic ventilation rate investigated, 1.5 c.f.m. per square foot of ceiling area, reduced the ceiling heat gain 43% at design conditions. This reduction was only 8% of the total residence heat gain. The maximum effective ventilation rate in Research Residence No. 2 would have been approximately 2 c.f.m. per square foot of ceiling area.

10. Good agreement was obtained between the measured temperature drop in ducts and the temperature drop calculated by the data presented in Engineering Experiment Station Bulletin No. 351.

11. Good agreement was obtained between the measured temperature rise in ducts and the temperature rise calculated by the method presented in the appendix of this bulletin. The appendix contains data for calculating the temperature rise in 4-, 5-, 6-, 7-, and 8-inch diameter ducts with 0, 1, or 2 inches of insulation with the ducts located in ambient temperatures from 75° to 150° F.

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The authors acknowledge the contributions of Mr. Herbert T. Gilkey and Mr. Chuan F. Chen to the investigations reported. Mr. Gilkey, formerly research associate in the Department of Mechanical Engineering in charge of the cooperative research program, supervised the preliminary planning of the investigation and aided in the analysis of the data collected during the 1954 cooling season and the 1954–55 heating season. Mr. Chen, formerly research assistant in the Department of Mechanical Engineering, aided in the collection and analysis of data during the 1954 and 1955 cooling seasons and the 1954–55 heating season.

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Checking of revisions was done by J. H. Healy, research associate in the Department of Mechanical Engineering.

The authors also wish to acknowledge the cooperation of the manufacturers that supplied equipment used in the investigation.

CONTENTS

CONTENTS (Continued)

FIGURES

FIGURES (Continued)

TABLES

I. INTRODUCTION

A. PRELIMINARY STATEMENT

This bulletin is a report of the results obtained during the 1954, 1955, and 1957 cooling seasons and the 1954–55, 1955–56, and 1957–58 heating seasons in Warm Air Heating Research Residence No. 2. This residence was completed in June 1947 and was built, furnished, and specifically equipped for conducting research in warm air heating and air conditioning by the National Warm Air Heating and Air Conditioning Association. It replaced the original Warm Air Heating Research Residence[1]* in which investigations of warm air heating and summer cooling systems were conducted from 1924 to 1946. The results of the research conducted during the years 1947–54 in Research Residence No. 2 have been reported in Engineering Experiment Station Bulletin No. 401, 427, and 445.[2, 3, 4]

This investigation was conducted under the terms of a cooperative agreement made in 1918 between the Association and the University of Illinois Engineering Experiment Station. In this cooperative research agreement the Association is represented by its Research Advisory Council. During the period of investigation herein reported 28 men served as members of the Council:

F. L. Meyer, Chairman (until January, 1957); Meyer Furnace Company, Peoria, Illinois.

K. T. Davis, Chairman (after January, 1957); Bryant Manufacturing Company, Indianapolis, Indiana.

P. R. Achenbach, Heating, Air Conditioning and Refrigeration Section, National Bureau of Standards, Department of Commerce, Washington, D.C.

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G. W. Denges, The Williamson Company, Cincinnati, Ohio.

R. S. Dill, National Bureau of Standards, Department of Commerce, Washington, D.C. (Deceased).

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W. E. Hood, Carrier Corporation, Syracuse, New York.

W. W. Johns, Johns and Son Furnace Company, Urbana, Illinois.

F. Lynn, Lennox Furnace Company, Marshalltown, Iowa.

W. M. Myler, Jr., Janitrol Heating and Air Conditioning Division, Surface Combustion Corporation, Columbus, Ohio.

C. W. Nessell, Minneapolis-Honeywell Regulator Company, Chicago, Illinois.

J. W. Norris, Lennox Furnace Company, Marshalltown, Iowa.

F. J. Nunlist, Worthington Corporation, Harrison, New Jersey.

N. A. Palmer, Eureka-Williams Company, Division of Henney Motor Company, Inc., Bloomington, Illinois.

G. Peoples, Lennox Furnace Company, Marshalltown, Iowa.

M. E. Ralston, The Williamson Company, Cincinnati, Ohio.

H. F. Randolph, International Heater Company, Utica, New York.

* Exponent numerals refer to corresponding entries in References Cited.

O. J. Ress, Mueller Climatrol Division of Worthington Corporation, Milwaukee, Wisconsin.

C. F. Suesserott, General Electric Company, Tyler, Texas.

W. H. Wentling, The Lau Blower Company, Dayton, Ohio.

H. Weyenberg, Holland Furnace Company, Holland, Michigan.

B. SCOPE OF INVESTIGATION

Many problems have arisen in the relatively new area of year-round air conditioning of residences, some of the most prominent of which are: differences in recommended air-flow rates for heating and cooling, different optimum locations and types of supply outlets, and differences in the relative heating and cooling load requirements for specific areas. The investigations in Research Residence No. 2 were conducted with the intent of solving these urgent problems and others and to determine the results to be anticipated from a central air conditioning system consisting of a warm air furnace and mechanical refrigeration unit when operated under specified conditions.

Three types of supply outlets were used during the investigations reported in this bulletin; perimeter floor diffusers, ceiling diffusers, and floor registers. Two types of mechanical refrigeration units were used in conjunction with a gas-fired furnace. One utilized water from the city mains to cool the condenser; the other unit, which was similar to the water-cooled unit and was made by the same manufacturer, utilized outdoor air to cool the condenser. Operation of these two units is compared and includes an analysis of operating costs.

It is now a widely accepted conclusion that perimeter diffusers are the best type of outlets for heating a residence by means of a forced warm air system. A recent bulletin[4] of the University of Illinois Engineering Experiment Station has reported the effectiveness of a perimeter system used for heating Research Residence No. 2. It is the purpose of this bulletin to compare performance of the perimeter diffuser system with the performance of systems utilizing floor registers, ceiling diffusers, and high sidewall registers.

Ceiling diffusers have become more popular in recent years, especially in areas where cooling is of primary importance. The installation of ceiling diffusers and attic-located air conditioning units is preferred by many building contractors since the ducts are located in attic spaces and may be installed after construction is completed. Home owners may prefer ceiling diffusers because they do not interfere with placement of furniture, floor coverings, or draperies.

Many homes are now undergoing modernization of their heating plants. Because a large number of these older homes have been heated by gravity warm air systems utilizing floor registers as the supply outlets, a considerable saving may result from utilization of these outlets with the modernized heating and/or cooling system.

The American Gas Association sponsored an investigation of the air distribution performance characteristics of various supply outlet types and locations and return intake locations, and the results of this investigation are reported in two University of Illinois Engineering Experiment Station bulletins,[5, 6] each entitled "Distribution of Air Within a Room for Year-Round Air Conditioning." Part I is contained in Bulletin No. 435 and Part II is contained in Bulletin No. 442. It was reported that ceiling diffusers which directed the air horizontally at the ceiling level produced more comfortable living conditions during cooling than did any other type of outlet, and if ceiling diffusers are utilized for heating the supply air temperature should not exceed 100° F. to maintain acceptable comfort conditions. It was also reported that outlets (such as floor registers), which provide vertical non-spreading air jets, are particularly applicable to a year-around system which utilizes a high air-flow rate and consequent low temperature differential between supply air and room air. Furthermore, with vertical non-spreading jets the number of outlets may be small, usually one per room, and they may be located near inside walls to utilize a simpler and less expensive duct system.

The room air distribution investigation was conducted in a laboratory test room under steady-state near-design conditions (usually 20° F. temperature difference across exposed wall during cooling and 75° F. temperature difference across the exposed wall during heating). The investigation in Research Residence No. 2 was conducted to study the two systems under actual operating conditions in an occupied residence with a varying load resulting from operation over a wide range of indoor-outdoor temperature differences.

The over-all objective of the investigations conducted in Research Residence No. 2 was to make thorough studies of the performance characteristics of the three year-around air conditioning systems and to compare the performance characteristics with those of a high sidewall system, with emphasis on evaluation of the comfort produced.

The specific objectives of this investigation were (a) to study the effect of air-flow rate and the accompanying supply air temperature on the comfort conditions produced by each system during both heating and cooling; (b) to study under- or over-cooling of certain areas when a system which has been adjusted for good room-to-room temperature balance during heating is subsequently used for cooling, and vice versa; (c) to study, during heating, the effect of excess furnace capacity (limited to a comparison at two fuel-input rates with the floor register system); (d) to compare the cost of operation of an air-cooled mechanical refrigeration unit with a waste-water-cooled unit; (e) to study the effect of night-air cooling on operating cost and comfort conditions; (f) to study attic ventilation as a means of reducing the ceiling heat gain.

A supplementary objective was to study the heat gain to and heat loss from ducts located in basements and attic spaces and to compare the gains and losses with calculated values.

The investigations herein reported consisted of studies of four separate and distinct systems and the results and conclusions are presented so that comparisons can readily be made between systems. Three sections are devoted to summaries of the comfort conditions produced by and operating characteristics of each outlet type with a fourth section containing a summary of the four systems; a section is devoted to room-to-room temperature balance; a section contains a comparison of air- and water-cooled condensing; a section is devoted to discussion of the effect of night-air cooling, and a section is devoted to discussion of forced attic ventilation to reduce the cooling load. In addition, the appendix contains calculated values of the heat gain to ducts located in spaces with various ambient conditions and with varying thicknesses of insulation and two types of vapor barriers.

C. GLOSSARY

Air-flow rate — The rate of circulation of air in cubic feet per minute (c.f.m.). Unless otherwise stated, all "c.f.m." values are for standard air density of 0.075 $\frac{\text{lb}}{\text{ft}^3}$.

Average surface temperature (A.S.T.) — The average surface temperature of the combined walls, ceiling, and floor, prorated on an area basis.

Balance of room air temperatures — Temperature difference between rooms, based on temperatures measured at the 30-inch level.

Bonnet — A plenum mounted at the discharge of an air conditioning unit.

Bonnet capacity — The heat output of the furnace available at the bonnet, B.t.u.h.

Bonnet efficiency — The ratio of the bonnet capacity to the heat liberated in the furnace by combustion of the fuel, expressed as a percentage. For gas-fired forced-air furnaces approved by the American Gas Association, the rated bonnet efficiency is 80%.

Breathing level temperature — Temperature of room air measured at a level 60 inches above the floor.

Ceiling diffusers — An outlet mounted in the ceiling which discharges the supply air so as to readily mix with room air.

Ceiling level temperature — Temperature of room air measured at a level 4 inches below the ceiling.

Combustion efficiency — The ratio of the heat liberated from the combustion of the fuel minus the stack, unburned combustibles, and excess air losses to the heat liberated, expressed as a percentage.

Datum temperature — During cooling, the mean daily temperature above which cooling unit operation occurs.

Drop zone — The area where the supply air enters the living zone during cooling.

Duct transmission efficiency — The ratio of the register delivery to the bonnet capacity, expressed as a percentage.

Extended plenum — A trunk duct that is uniform in size along its entire length.

Floor diffusers — A floor outlet in which the front vanes are deflected progressively from vertical at the center to inclined at the ends to produce a fan-shaped air discharge pattern.

Floor level temperature — Temperature of room air measured at a level 4 inches above the floor.

Floor register — A floor outlet in which the front vanes are all vertical to produce a nonspreading air discharge pattern.

Forced attic ventilation — Ventilation of an attic space with outdoor air by means of a fan.

High sidewall register — An outlet located high on the inside walls. Usually, the front vanes are vertical and deflect the supply air in two horizontal directions.

Indoor-outdoor temperature difference — The difference in temperature between indoor air and outdoor air. Large temperature differences indicate severe weather, small differences indicate mild weather.

Isotherm, floor surface — Lines indicating points of equal temperature.

Isovel — Lines indicating points of equal velocity.

Living zone — The space in a room between the floor level and the breathing level.

Mean daily temperature (M.D.T.) — The mean of the maximum temperature occurring in a 24-hour period and the minimum temperature occurring during the previous night. Used as an index of the daily average performance characteristics during cooling.

Mean radiant temperature (M.R.T.) — The effective temperature of all surfaces which are in radiant heat interchange with a particular surface.

Night-air cooling—An operating procedure wherein windows and doors are opened during cool summer nights.

Perimeter system — A forced-air system in which the conditioned air is supplied to the rooms from diffusers located near areas of high load, such as windows and exposed walls.

Register delivery — The air conditioning capacity available at the outlets, B.t.u.h. This is based on the air-flow rate and the difference between room air and supply air temperature.

Return air intake (grille) — An opening connected by ductwork to the return air side of the air conditioning unit to supply room air to be conditioned.

Sitting level temperature — Temperature of room air measured at a level 30 inches above the floor.

Supply (face) velocity — The velocity obtained by dividing the air-flow rate through an outlet by the free (minimum passage) area, in feet per minute (f.p.m.).

Temperature variation, room air — The variation in room air temperature between two elevations. In this report the sitting level is considered as the reference level.

Temperature gradient — A graphical representation of air temperatures occurring at various levels in a room at one location.

Thermocouple — A device consisting of two small wires of dissimilar metals, (usually copper and constantan) for measuring temperature.

Thermostat, anticipating — A thermostat with a small electric heater mounted within its casing to alter the thermostat differential.

Thermostat differential — The fluctuation of room air temperature at the thermostat.

Throw — The distance from an outlet at which the supply air velocity decreased to a specified value, usually 50 f.p.m.

Year-around air conditioner — A device consisting of a furnace and refrigeration unit used to change the temperature, humidity, and dust content of air used to air-condition a space.

II. DESCRIPTION OF EQUIPMENT AND PROCEDURE

A. RESEARCH RESIDENCE NO. 2

The residence, which is shown in Figure 1, was a one-story structure of frame construction and had a full basement. A detailed description of the residence has been presented in Engineering Experiment Station Bulletin No. 401.(2) A summary of the construction data, room dimensions, calculated heat gain, and calculated heat loss is presented in Table 1, and a floor plan of the residence is shown in Figure 3. The calculated heat gain of the first-story rooms for design temperatures of 95° F. outdoor-air and 75° F. indoor-air and a medium daily outdoor-air temperature range, calculated in accordance with *Manual 11* (7) of the National Warm Air Heating and Air Conditioning Association, was 22,605 B.t.u.h. The calculated heat loss of the first-story rooms for design temperatures of −10° F. outdoor-air and 70° F. indoor-air temperature, calculated in accordance with *Manual 3*(8) of the National Warm Air Heating and Air Conditioning Association, was 31,047 B.t.u.h.

The residence contained some special construction features, one of which was open-web steel ceiling and floor joists. In an earlier heating investigation(1) a warm-air panel was constructed in the ceiling but for the investigation reported herein mineral wool batt-type insulation 3⅝-inch thick was placed between the joists and an additional 2-inch thickness of loose-fill-type mineral wool insulation was spread over the batts to offset the high thermal conductivity of the steel joists. The

Figure 1. Warm Air Heating Research Residence No. 2

over-all coefficient of heat transmission of the ceiling was 0.07 B.t.u.h. per square foot (°F.).

The floor was constructed of 2½-inch thick gypsum planks supported by the open-web steel joists. Pressed cement-asbestos sheets were placed on top of the gypsum planks and the finish floor was asphalt tile.

The residence was occupied by a family of two adults during all studies except the 1957–58 heating season when it was occupied by a family of two adults and one infant. All observations were made under normal living conditions.

B. YEAR-AROUND AIR CONDITIONING SYSTEM

1. Year-Around Air Conditioner

The year-around air conditioner, which was located in the basement, consisted of two matched packages, one containing the mechanical refrigera-

Table 1

Data on Research Residence No. 2

A. Heat Transmission Coefficients, B.t.u.h. per sq. ft. (°F.)	U
Insulated Frame Wall, with 3⅝-inch mineral wool insulation	0.07
Insulated Ceiling, mineral wool insulation	0.07
Outside Doors, equipped with storm doors (Heating)	0.45
Outside Doors, equipped with screens only (Cooling)	1.13
Windows, equipped with storm sash (Heating)	0.45
Windows, equipped with screens only (Cooling)	1.13
Fixed Window in Living Room, double glass	0.06
B. Infiltration Factors, cu. ft. per hr. (feet of crack)	I
Door, weatherstripped and equipped with storm door (Heating)	55
Door, weatherstripped and equipped with screen only (Cooling)	138
Window, weatherstripped	24
Fixed Window in Living Room	14

C. Room Dimensions, Floor Area, Volume, and Design Heat Loss and Heat Gain

Room	Internal Dimensions				Floor Area, ft.²	Volume,[a] ft.³	Design[b] Heat Gain, B.t.u.h.		Design[c] Heat Loss, B.t.u.h.
	ft.	in. x	ft.	in.			Sensible	Total	
Living Room	21	10	13	4	292.0	2,480	5,835	7,580	9,510
South Bedroom and South Bedroom Closets (2)	13	4	11	0	155.5	1,325	3,480	4,520	5,598
	4	6	1	11	17.4	147	...[d]	...[d]	...[d]
Bath	7	8	4	11	38.0	320	980	1,280	1,453
North Bedroom	11	11	10	4	123.3	1,050	3,400	4,420	5,620
North Bedroom Closet	5	10	2	4	13.6	116	...[d]	...[d]	...[d]
Hall to Bath	6	7	5	2	34.0	289	...[d]	...[d]	...[d]
Front Hall	11	6	4	7	52.6	448	...[d]	...[d]	2,826
Front Hall Closet	4	0	2	4	9.3	79	...[d]	...[d]	...[d]
Kitchen-Dinette	19	0	11	4	215.0	1,830	3,695	4,805	6,040
TOTAL					950.7	8,084	17,390	22,605	31,047

[a] Room volumes based on internal dimensions of rooms and include cabinets and closets.

[b] Heat gain calculations based on third edition[7] of *Manual 11* for indoor air temperature of 75° F., outdoor air temperature of 95° F., and a medium daily outdoor air temperature range.

[c] Heat loss calculations based on third edition[8] of *Manual 3* for indoor air temperature of 70° F. and outdoor air temperature of −10° F.

[d] Heat loss and heat gain for these rooms included with larger adjoining rooms.

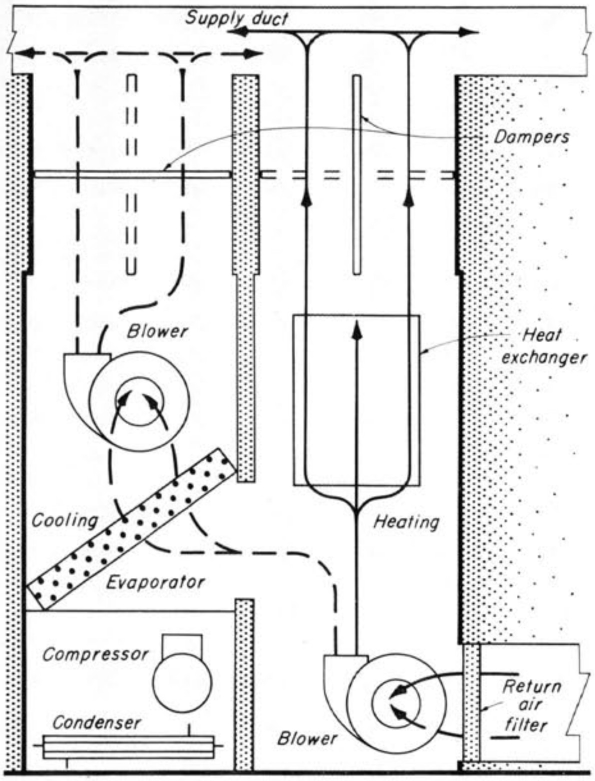


Figure 2. Schematic diagram of year-around air conditioning unit

tion unit and the other containing the forced warm-air furnace. The cooling and heating packages were arranged for parallel air flow so that the recirculated air passed through either one unit or the other, depending upon the season. A schematic sketch of the air conditioner, with the alternate air-flow paths indicated, is shown in Figure 2. During heating, the return air entered the base of the furnace through a replaceable viscous impingement-type air filter, was drawn through the furnace blower, and then passed vertically into and through the heat exchanger and was discharged into the bonnet mounted atop the furnace. During cooling, the return air also entered the base of the furnace through the filter and was drawn through the furnace blower, but then the air passed horizontally through an opening between the furnace and cooling unit. The air entered the cooling unit beneath the inclined evaporator coil, passed through the cooling unit blower and was subsequently discharged into the bonnet mounted atop the cooling unit. Therefore, during heating only the blower located in the furnace was used for recirculating air; and during cooling, because of the increased resistance to air flow imposed by the coil, the two blowers were used in series.

The change of the air-flow path from heating to cooling operation or vice versa was accomplished by opening and closing the appropriate dampers located in the cooling and heating bonnets, as shown in Figure 2.

The furnace was gas fired, of the hi-boy type, and had a rated input of 70,000 B.t.u.h. The furnace was equipped with an integrally mounted forward curved blade centrifugal blower which had a 9-inch-diameter wheel and was powered by a 110-volt single phase motor. Natural gas with a heating value of 1000 B.t.u.h. per cubic foot was the fuel used. The fuel input rate was adjusted to conform with the calculated heat losses by interchanging burner orifices and adjusting the burner manifold pressure regulator.

The condenser of the cooling unit indicated in Figure 2 was cooled by water from the city water supply and the water was subsequently discharged to the sewer. An automatic regulating valve controlled the flow of water to the double tube type condenser. The hermetically sealed compressor was driven by a 230-volt, single-phase, 2-h.p. motor which was cooled by the suction gases. The finned-tube evaporator had a face area of 1.86 square feet and was constructed of $\frac{1}{2}$-inch copper tubes spaced three rows deep and nine rows wide. The evaporator tubes and the aluminum fins had been bonded by hydraulically expanding the tubes with the fins in position. The fins, which were $\frac{1}{2}$ inch deep at the face, were spaced 10 per inch along the tubes. The flow of refrigerant (Refrigerant 22) to the two sections of the coil was throttled by capillary tubes. A forward curved blade centrifugal blower with a 10-inch diameter wheel was mounted above the evaporator coil. This blower was powered by a 110-volt single-phase motor. The cooling unit had a rated capacity of 24,088 B.t.u.h. with an 800-c.f.m. air-flow rate through the evaporator at ASRE specified inlet air temperatures and with condenser water temperatures of 75° F. and 95° F. at the inlet and outlet, respectively.

2. Controls

The temperature of the conditioned space was controlled by a thermostat located at the 30-inch level in the entrance hall. During heating, the thermostat was an anticipating type, and during cooling the thermostat was nonanticipating. The thermostat had a differential adjustment which was maintained at the minimum setting. Only the furnace recirculating blower was used during heating. The recirculation blower was controlled by a thermostatic-type control mounted on the furnace casing with its element in the heated air stream during all heating test series except one when the blower was set for continuous operation. The blower was set to cut on at 100° F. supply air temperature and to cut off at 80° F. supply air temperature when set for cyclic operation. The furnace and cooling unit blowers were both operated during cooling. During one cooling test series the blower was operated during the compressor "on" periods only and during all other cooling test series the blower was operated continuously. The furnace was equipped with a limit switch which was set to close the furnace gas valve if the supply air temperature attained a temperature of 200° F. and to reopen the gas valve after the supply air temperature had dropped to 175° F.

3. Supply Outlets

The perimeter floor diffusers were 2¼ inches by 14 inches and had vanes across the shorter dimension. The angle of the vanes varied from vertical at the center of the 14-inch side of the diffuser to approximately 22½ degrees from vertical at each end.

The ceiling diffusers were circular, two-cone, two-position diffusers with the cones positioned to produce a horizontal air discharge pattern. The diffusers were located near the center of the room.

The floor registers had vertical vanes and were equipped with multiple-leaf dampers which were in the wide-open position throughout the investigation. The high sidewall system, which was studied during a previous investigation, utilized registers located on the inside walls 6½ feet above the floor. The register vertical vanes were set to direct the air at angles of 22 degrees to the right and left of the vertical centerline of the register.

4. Return Grille

With each of the three types of supply outlets listed above, a single, centrally located return air grille was used. The grille was located in the floor of the entrance hall.

C. INSTRUMENTATION

Approximately 250 copper-constantan thermocouples were used to measure temperatures. Thermocouples for measuring room air temperatures at four levels were installed on standards located near the center of the first-story rooms and at three locations in the basement. Thermocouples were also installed on ceiling, wall, and floor surfaces, in the attic, in the duct systems, and at other locations inside and outside the residence. All thermocouples were connected to switch panels located in the basement and the switch panels were in turn connected to an indicating potentiometer. Recording potentiometers provided a means of continuously recording temperatures at 24 selected locations. Heat-flow meters were installed in each wall section, on the underside of each roof section, on the inside surface of each gable, and beneath the insulation in the ceiling of each room.

A thermo-humidigraph was located at floor level near the thermostat in the entrance hall to record the relative humidity of the conditioned space. The thermo-humidigraph was periodically calibrated by an aspirated psychrometer. The recording and indicating potentiometers were periodically calibrated by means of a precision Wheatstone bridge potentiometer.

During heating, the flue gas temperature at the furnace combustion chamber outlet was recorded continuously by a recording thermometer. The CO_2 content of the flue gas at the furnace outlet was periodically measured by means of a CO_2 analyzer which sampled the flue gas ahead of the draft hood. Self-starting electric clocks were used to obtain the operating time of the burner and blower, and a complete record of the number and frequency of burner and blower operations was obtained by electrically actuated operation pens. The electrical energy consumed by the furnace blower and the total electrical energy consumed for all purposes were separately metered. The gas consumed by the furnace was metered separately from the gas used for cooking and water heating.

During cooling, the condenser water flow rate was recorded by means of an orifice differential pressure recorder, the water inlet and outlet temperatures were recorded on the indicating potentiometers, and the total water used daily for cooling the condenser was metered separately from the total water used daily in the residence. Self-starting electric clocks were used to obtain the total compressor operating time and electrically actuated operation pens recorded the number and frequency of compressor operations. The electrical energy consumed by the compressor motor, by the recirculation blower motors, and the total electrical energy consumed in the residence were metered separately.

During both heating and cooling the air-flow rate for each system was determined by means of a vane-type anemometer installed in the return duct and calibrated in place.[2]

Records of solar radiation received on a horizontal surface were obtained by means of a solar pyriheliometer and an electronic recording potentiometer which operated from sunrise to sunset. Wind speed was continuously recorded and the wind direction was noted twice each day.

D. GENERAL OPERATING PROCEDURE

Either continuous or periodic records were obtained of all significant temperatures. Complete daily records were kept of the operating time, the number of operation cycles, the electrical consumption of the blower motor(s), the compressor, and the total electrical consumption of the residence. The gas consumed by the furnace and for household purposes was recorded daily.

Because the most critical period of operation of an air conditioning system is at the time of maximum load, the test days were set up accordingly. The test days began near the time of minimum load, which during cooling is in the early morning and during heating is near mid-day. The cooling test day began at 5:30 a.m. and continued until 5:30 a.m. the following day. The heating test day began at 11:30 a.m. and continued until 11:30 a.m. the following day. All daily electrical energy, water, gas, and operation time data are based on these test days. A complete record of temperature was obtained at approximately the time of maximum load, which during heating was at 6:30 a.m. and during cooling was at 3:00 p.m. All times are central standard time.

For the purpose of making comparisons, studies were conducted for periods of from two to several weeks, thus obtaining the performance characteristics of each system under the specific test conditions being investigated over a range of weather conditions.

Comfort depends upon a large number of factors, including air temperatures, surface temperatures, relative humidity, and air movement, as well as more subjective sensations such as odor, noise, and dust content. It was not possible to evaluate all the items enumerated above and, therefore, emphasis was placed on air and surface temperatures and measurements were made of air movement and relative humidity. No attempt was made to evaluate odor, noise, or dust content of the air.

III. PERFORMANCE OF SYSTEMS DURING HEATING

A. SYSTEMS INVESTIGATED

The supply outlet locations and duct systems for the three air distribution systems which were investigated and the high sidewall system investigated previously are shown in Figures 3, 4, 5, and 6. Nine $2\frac{1}{4}$-inch by 14-inch floor diffusers were used with the perimeter air distribution system. The diffusers were located below windows except in the bath and kitchen where such locations were impractical. The duct work consisted of extended plenums and 4-inch-diameter branch ducts. The east extended plenum was 8 inches by 8 inches and the west extended plenum was 14 inches by 8 inches. Each branch duct was equipped with a volume damper for adjusting the air-flow rate. All duct work was uninsulated.

The ceiling diffusers were circular, two-cone, two-position diffusers with the cones positioned to produce a horizontal discharge air pattern. A diffuser was located near the center of each room except in the living room where conditioned air was supplied through two diffusers, with one located at the center of each of two equal areas. The diffusers had 6-, 8-, and 10-inch-diameter collars (diffuser sizes are listed according to collar diameter). The diffusers were selected so that each had an average throw equal to the distance to the adjacent wall, according to the manufacturer's catalog data, based upon a total air-flow rate of 800 c.f.m. apportioned according to the calculated heat gain. The duct work used to supply conditioned air to the ceiling diffusers was located in the attic. The bonnet of the year-around conditioner, which was located in the basement, was connected to an attic plenum by a 9-inch by 18-inch duct enclosed in a corner of the dinette. Individual circular sheet metal ducts extended from the attic plenum to each ceiling diffuser. These branch ducts were of 5-, 6-, and 7-inch diameter, and each was equipped with a volume damper to provide for the adjustment to the air-flow rate. The collar diameters of all ceiling diffusers, except the 6-inch diffuser located in the kitchen, were larger than the corresponding branch ducts and, therefore, tapered transitions with included angles of approximately 15 degrees were used to connect the collars to the ducts.

All duct work in the attic, including the plenum, was insulated with 2-inch-thick glass fiber insulation enclosed in a vapor barrier. The vapor barrier used on three of the branch ducts was aluminum foil and on the other four branch ducts was duplex paper.

The floor registers had vertical vanes and were equipped with multiple-leaf dampers which were in the wide-open position during the investigation. A register was located in the floor near the center of an inside wall in each of the six rooms of the residence. The registers ranged in size from $2\frac{1}{4}$ inches by 14 inches to 8 inches by 12 inches. All outlet dampers remained fully open throughout the investigation. The register sizes were determined on the basis of free area required to provide a supply velocity of 500 f.p.m. with a total air-flow rate of 800 c.f.m. apportioned according to the calculated heat gain. The duct work used to supply conditioned air to the registers consisted of rectangular extended plenums and relatively short branch ducts. The east extended plenum was 10 inches by 8 inches and the west extended plenum was 14 inches by 8 inches. The branch ducts ranged in size from 5-inch diameter to 5-inch by 12-inch rectangular. Each branch duct was equipped with a volume damper for the adjustment of the air-flow rate. The duct work was uninsulated.

The same return air grille and return air duct system were used with the perimeter diffusers, floor registers, and ceiling diffuser systems. The return air grille was located in the entrance hall and is shown on the floor plans in Figures 3, 4, 5, and 6. The grille consisted of a 24-inch by 8-inch floor grille combined with a 24-inch by 6-inch baseboard grille which opened into a 4-inch stud space. The total free area of this combination was 289 square inches. The 24-inch by 12-inch opening in the floor (8-inch wide floor grille opening plus stud space) was reduced to 16 inches by 12 inches by means of

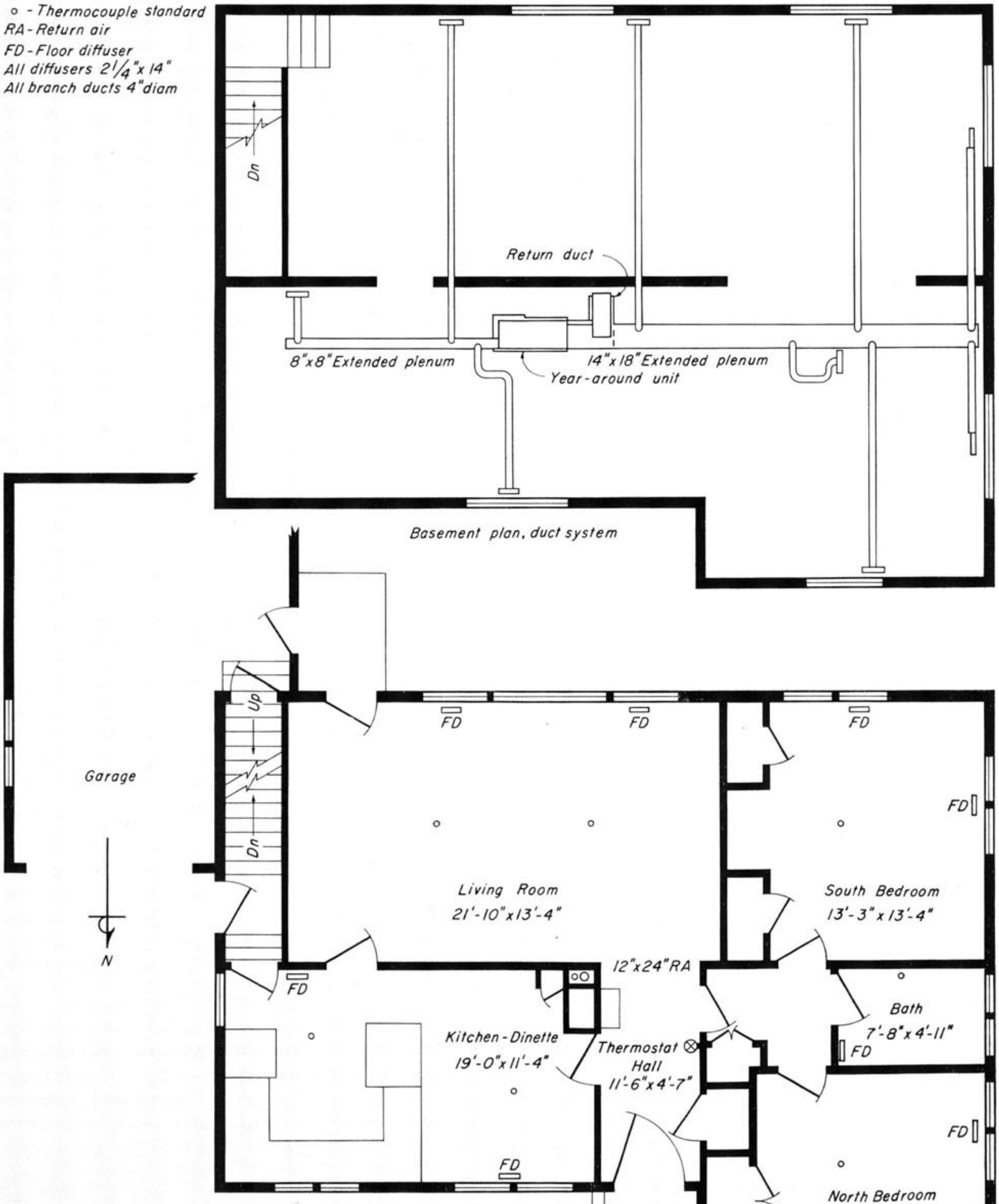


Figure 3. Floor diffuser locations

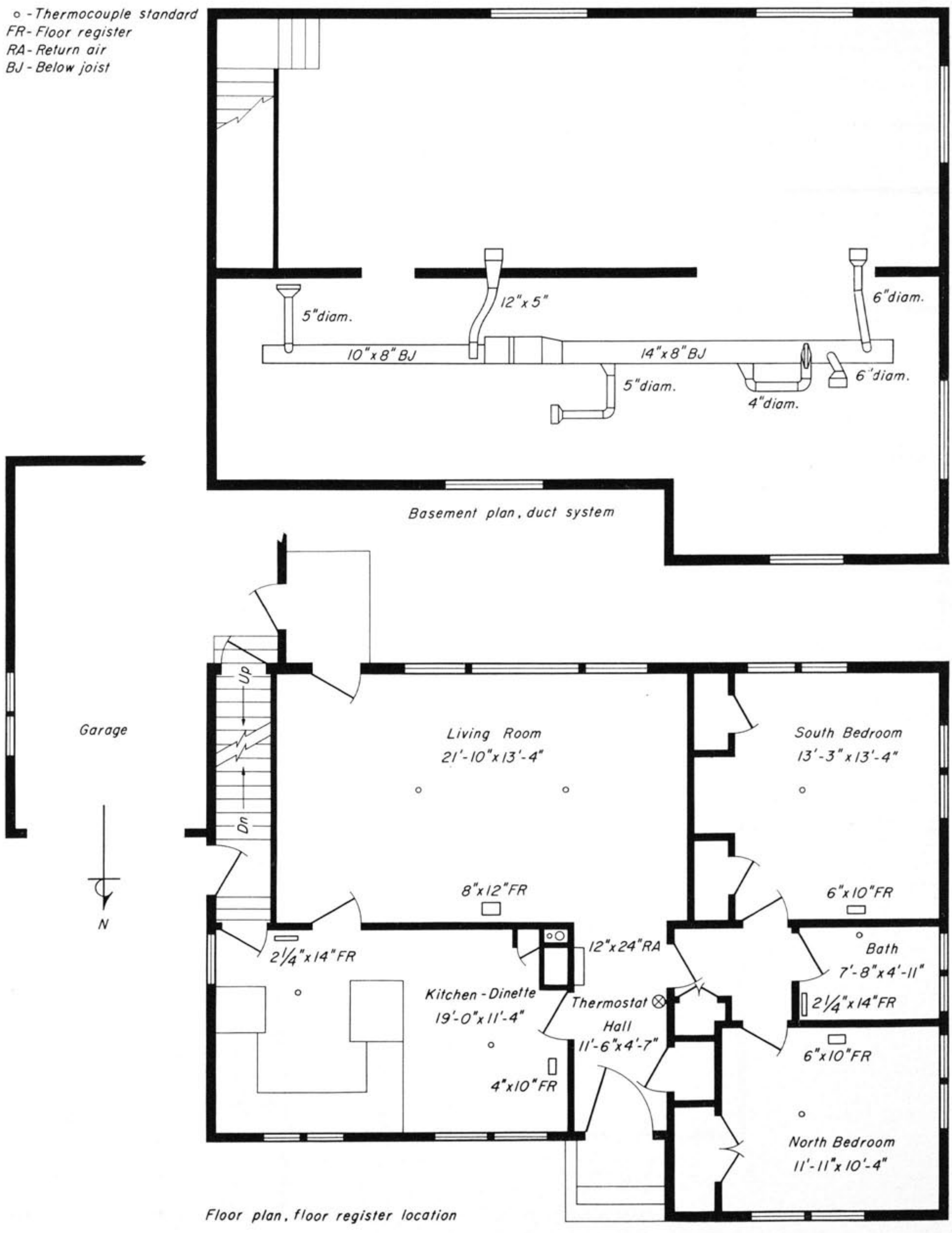


Figure 4. Floor register locations

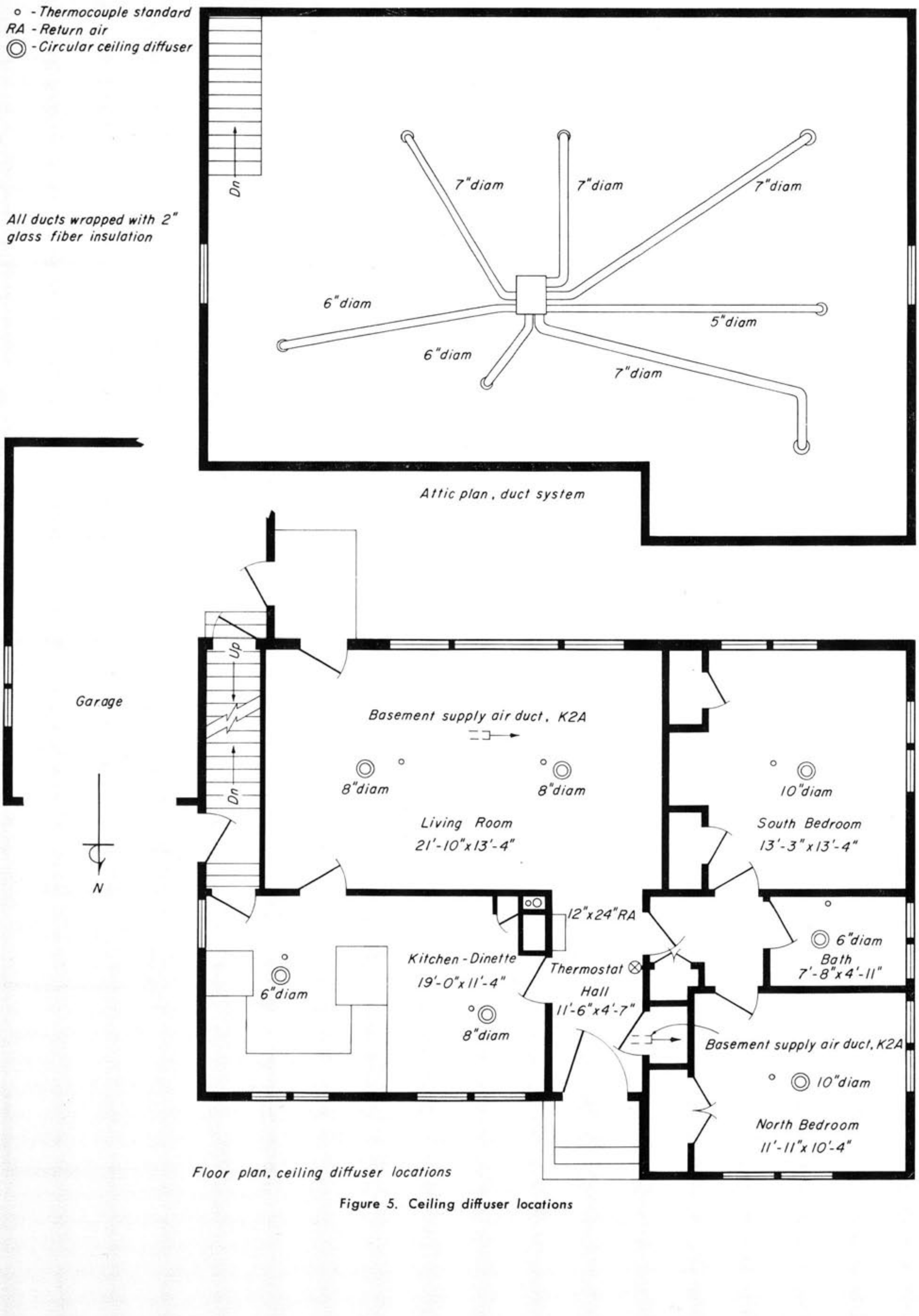


Figure 5. Ceiling diffuser locations

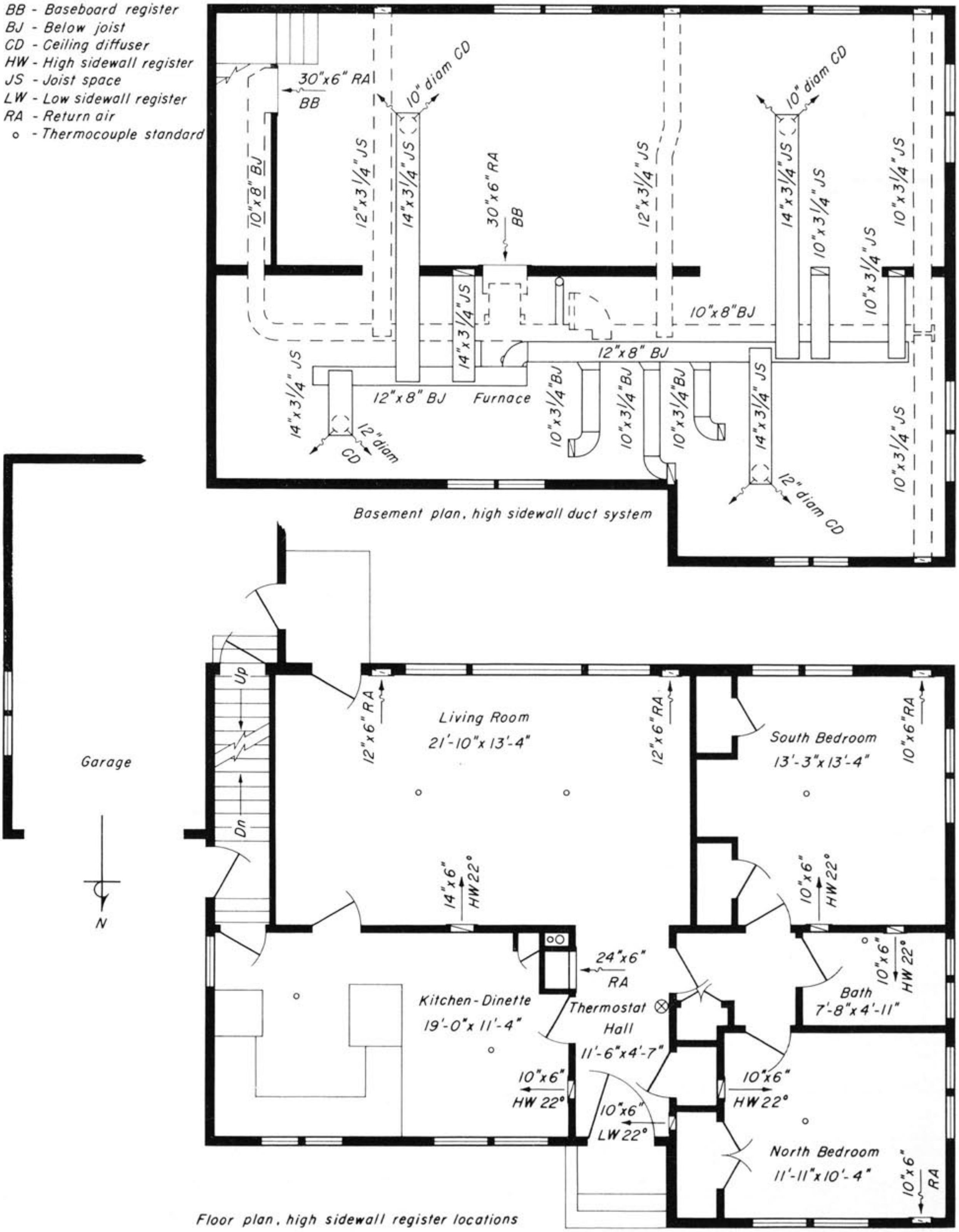


Figure 6. High sidewall register locations

Table 2

Series Designations and Controlled Variables During Heating

Floor Diffusers*				Ceiling Diffusers				Floor Registers			
Series	System Air-Flow Rate, c.f.m.	Fuel-Input Rate, B.t.u.h.	Calculated Air Temperature Rise, °F.	Series	System Air-Flow Rate, c.f.m.	Fuel-Input Rate, B.t.u.h.	Calculated Air Temperature Rise, °F.[c]	Series	System Air-Flow Rate, c.f.m.	Fuel-Input Rate, B.t.u.h.	Calculated Air Temperature Rise, °F.[c]
1954–55 Heating				*1957–58 Heating*				*1957–58 Heating*			
G-1	530	45,660	69	K-1	800	45,660	42	L-1	800	45,660	42
G-2	475	"	77	K-2	600	45,660	56	L-2	600	45,660	56
G-3	295	"	124	K-2A[b]	793	60,360	56	L-3	400	45,660	84
G-4	295	"	124	K-3	400	45,660	84	L-4[d]	800	70,000	65
1955–56 Heating											
G-11[a]	530	"	69								
G-12[a]	420	"	87								
G-13[a]	420	"	87								

* Fuel-input rate, 45,660 B.t.u.h.
[a] During the 1955–56 heating season an auxiliary electric heater was used in the living room to simulate a reduced ratio of heat loss to heat gain in order to study an extreme case of temperature unbalance.
[b] Heat added below the floor. 600 c.f.m. to first story, 193 c.f.m. to basement.
[c] Based on 80% furnace efficiency.
[d] Fuel-input rate set at rated capacity of furnace.

a tapered transition, and the return duct was of constant dimensions from that point to the return plenum.

Extended plenum ducts were used with the high sidewall system. The branch ducts were connected to the top or side of the trunks and were unchanged in size from the trunk take-off fitting to the register stackhead. All registers in the first-story rooms were at the high sidewall location, 6½ feet from the floor, with the exception of the baseboard register in the front hall near the door. The return air intake for the first story was a 24-inch by 8-inch baseboard grille located in the front hall. When the basement was heated, ceiling outlets of the circular-diffuser type were used to distribute warm air to the basement. The basement return air intake was located in the baseboard.

B. EXPERIMENTAL CONDITIONS

Table 2 is a summary of the experimental conditions and the controlled variables used with the three air distribution systems. The conditioned space consisted of all first-story rooms. The basement was heated only by losses from the furnace, furnace vent pipe, duct work, and heat-producing appliances except as noted under Series K-2A where heat was added below the first-story floor. The fuel-input rate was set at 45,660 B.t.u.h., which was equal to the first-story heat loss divided by the product of the assumed bonnet and duct transmission efficiencies: $\frac{31{,}047}{0.80 \times 0.85}$. No difficulty from flame failure was experienced although the fuel-input rate was considerably less than the 70,000 B.t.u.h normal rated input of the furnace. The thermostat was adjusted to maintain a temperature of approximately 72° F. at the 30-inch level.

The small-pipe system with floor diffusers was under investigation during the 1954–55 and 1955–56 heating seasons, during which seven series of studies were conducted. Series G-1 and G-11 were for the purpose of studying temperature balance during heating, utilizing cooling air-flow rates, with the system balanced for summer cooling. Series G-2 and G-12 also utilized the cooling air-flow rate, minus the air-flow to those rooms where ducts were shut off to improve the temperature balance during heating. Series G-13 was for the purpose of comparing the effects of continuous and cyclic blower operation on the temperature balance. The auxiliary electric heater in the living room during Series G-12 and G-13 was used to simulate a room which required a considerably larger portion of the total air-flow rate for cooling than it did for heating.

The calculated air-flow rate required for 100° F. air temperature rise with the furnace input set at 45,660 B.t.u.h. was 340 c.f.m.; therefore, this was the air-flow rate necessary to conform to the principle of continuous air circulation as outlined in the 1947 edition of *Manual 6*.[9] Because the furnace was fired at only 65% of its rated capacity, its efficiency was less than 80% and the air-flow rate of 340 c.f.m. did not result in a 100° F. air temperature rise. With the air-flow rate reduced to 295 c.f.m., the measured air temperature rise was approximately 85° F. as compared with a calculated temperature rise of 100° F.

Four series were conducted with the ceiling diffuser system used during part of the 1957–58 heating season. The series designations and the controlled variables are listed in Table 2. Furnace

Table 3

Room Air Velocities at Sitting and Floor Levels in Living Room

Series	Outdoor Air Temperature, °F.	Average Supply Air Temperature, °F.	Level	Percentage of Traverse Points Having Velocities Within Listed Range: Below 15 f.p.m.	15 to 35 f.p.m.	Above 35 f.p.m.	Average Velocity f.p.m.	Maximum Velocity f.p.m.
Ceiling Diffusers								
K-1	11	97	Sitting	7	93	0	18	35
			Floor	100	0	0	8	11
K-2	28	95	Sitting	0	100	0	19	25
			Floor	3	97	0	15	20
K-3	22	118	Sitting	100	0	0	7	10
			Floor	100	0	0	5	10
Floor Registers								
L-1	10	92	Sitting	0	87	13	29	45
			Floor	0	87	13	30	45
L-2	10	98	Sitting	0	97	3	23	40
			Floor	90	10	0	12	15
L-3	10	103	Sitting	70	30	0	12	20
			Floor	100	0	0	8	10

fuel-input rate and air-flow rate with the accompanying air temperature rise were the controlled variables. During the three principle series, designated as Series K-1, K-2, and K-3, the fuel-input rate was set at 45,660 B.t.u.h. The fuel-input rate during Series K-2A was increased to 60,360 B.t.u.h. to determine the effect on floor surface temperatures of introducing 10,000 B.t.u.h. below the first-story floor. The calculated air-flow rate for a 90° F. air temperature rise was 377 c.f.m. with the furnace input adjusted to 45,660 B.t.u.h. This was the air-flow rate necessary to conform with the principle of comfort air circulation as outlined in the 1957 edition of *Manual 6*.[10] Since almost all year-around installations require a greater air-flow rate for cooling than that necessary for a 90° F. rise during heating, one objective of the 1957–58 investigation was to determine the effect on comfort of greater air-flow rates which would result in less than a 90° F. air temperature rise.

Four heating series were conducted with the floor register system during the 1957–58 season. As with the ceiling diffusers, the fuel-input rate and the air-flow rate with the accompanying air temperature rise were the controlled variables. Throughout Series L-1, L-2, and L-3, the fuel-input rate was set at 45,660 B.t.u.h. The Series L-4 fuel-input rate was increased to the rated value of 70,000 B.t.u.h. The air-flow rate was the controlled variable during the first three series, varying from the cooling air-flow rates of 800 and 600 c.f.m. during L-1 and L-2, respectively, to 400 c.f.m. during L-3. The Series L-4 air-flow rate was 800 c.f.m.

The high sidewall system studied during a previous investigation was operated in two ways, with the basement heated and also with the basement unheated. With the basement heated the fuel input was 76,000 B.t.u.h. and the 100° F. rise air-flow rate was 565 c.f.m. With no warm air introduced into the basement the fuel input rate was 46,000 B.t.u.h. and the 100° F. rise air-flow rate was 340 c.f.m. In both cases the fuel-input rate was equal to the calculated heat loss divided by 0.8 (bonnet efficiency of 80% assumed). The heat loss of the first-story rooms was 31,047 B.t.u.h. The combined heat loss of the first-story rooms and basement was 60,782 B.t.u.h.

C. UNIFORMITY OF ROOM AIR TEMPERATURES AT VARIOUS LEVELS AND ROOM AIR VELOCITIES

The supply air from the floor diffusers was projected vertically in a plane parallel to the wall. The vanes of floor diffusers are set in such a way that the discharge jet forms a fan shaped pattern that "blankets" the exposed wall. The spreading jet partially counteracts the cold currents which flow down the walls. From previous studies,[4] the variation in the living room at the floor level was 2° to 3° F. and less than 1° F. at the sitting level under conditions of average winter weather (approximately 30° F., wind less than 10 m.p.h.). Under conditions of lower temperatures and higher winds, the floor level variation increased by another 1° F. The minimum living room temperatures occurred near the door in the southeast corner where the exposed wall was not blanketed by the warm supply air.

Table 3 is a summary of the living room air velocities measured at 50 locations in the planes of the sitting and floor levels. For comparative purposes summaries are listed for both the ceiling diffusers and the floor register system. The velocities are divided into three ranges: below 15 f.p.m., 15 to 35 f.p.m., and above 35 f.p.m. Of the three, the

Table 4

Summary of Cyclical Temperature Variation Studies

Floor Diffusers

	Cyclical Variation[a], °F.	
	Continuous Blower[c]	Cyclic Blower
Outdoor Air Temperature, °F.	49	62
Location		
Thermostat	0.7	0.3
Liv. Rm. E.	0.9	0.8
Liv. Rm. W.	1.1	0.8
S. Bedroom	0.7	1.5
Bath	1.3	1.9
N. Bedroom	0.7	1.5
Dinette	0.9	1.5
Kitchen	0.7	1.1
Avg.	0.9	1.3

Ceiling Diffusers

				Cyclical Temperature Variation, °F.	
Series	Outdoor Air, °F.	Blower Operation	Air-Flow Rate c.f.m.	Room Air[b]	Thermostat
K-1	32	Cyclic	800	0.9	0.8
K-2	23	Cont.	600	0.4	0.2
K-3	30	Cont.	400	0.2	0.2

Floor Registers

					Cyclical Temperature Variation, °F.	
Study No.	Series	Outdoor Air, °F.	Blower Operation	Air-Flow Rate c.f.m.	Average Room Air[b]	Thermostat
1	L-1	35	Cyclic	800	2.1	0.9
2	L-1	35	Cont.[c]	800	0.9	0.3
3	L-2	19	Cont.	600	0.5	0.6
4	L-3	26	Cont.	400	0.3	0.3
5	L-4	32	Cyclic	800	2.3	1.2
6	L-4	32	Cont.[c]	800	1.4	0.7
7	L-4	32	Cyclic	800	2.4	1.1
8	L-4	52	Cont.[c]	800	1.6	0.9

[a] Air-flow rate 420 c.f.m. fuel-input rate; 45,660 B.t.u.h.
[b] Based on sitting level room air temperatures measured at the locations of the seven permanent thermocouple standards.
[c] Blower maintained "on" by manual setting of fan switch.

15 to 35 f.p.m. range is preferable. A region with motion less than 15 f.p.m. is considered stagnant and a region with velocities greater than 35 f.p.m. is considered drafty.

The ceiling diffusers produced low room air velocities within the living zone. There were no objectionable drafts. The maximum velocity during each series occurred near the return air grille and, in general, higher velocities occurred with the 600 and 800 c.f.m. air-flow rates used during Series K-1 and K-2, respectively. Temperatures were measured at the same locations as the velocities, and variations at either level were approximately the same with all of the air-flow rates. At the sitting level in the living room the air near the interior wall was 1° F. warmer than the air near the exposed wall. At the floor level, the variation from the interior to the exposed wall was 2° F. The average temperature at the floor level was 4° F. lower than the average sitting level temperature with an 11° F. outdoor air temperature.

The lowest projection of the supply air with the ceiling diffusers occurred in the north bedroom where the air velocity was 50 f.p.m. at a distance 6 feet above the floor. In the bath a 50-f.p.m. velocity was measured 7 feet above the floor. In other rooms the walls were beyond the radius of diffusion and the air was not projected down the walls.

The warm air from the floor registers, which were located next to the inside walls, was discharged in a confined vertical jet that spread over the ceiling. At the higher air-flow rates (Series L-1, 800 c.f.m.) the velocity at practically all locations, as is shown in Table 3, was in the acceptable range between 15 and 35 f.p.m. The maximum velocity of 55 f.p.m. was measured near the return air grille. The room air motion decreased with the air-flow rate and during Series L-3 (400 c.f.m.) the velocity at most locations was below 15 f.p.m. The temperature variation in the plane of the sitting level was approximately 1° F. during all series. The lower temperatures were measured near the exposed wall. With the higher air-flow rate the temperature variation at the floor level was also 1° F., but at the lower air-flow rates the variation increased to 3° F. The higher air-flow rate resulted in a 480-f.p.m. face velocity at the living room register which was sufficient to project the air across the ceiling and down the exposed wall to help prevent drafts due to air cooled in contact with the glass surfaces. This projection of supply air was also evidenced by a small area of warmer air at the sitting level near the exposed wall directly opposite the register.

D. CYCLICAL VARIATION OF ROOM AIR TEMPERATURES AT THE SITTING LEVEL

One measure of comfort is the variation in room air temperature that occurs during cyclic operation of the burner and blower. Cyclical variation of room air temperatures is generally least with continuous blower operation. This method of operation is most nearly approached, under all weather conditions, with the furnace adjusted to comply with the continuous air circulation (C–A–C)[9] principle. Table 4 is a summary of the studies of cyclical temperature variations at the sitting level which were conducted with each system. A room-by-room summary is shown for the floor diffusers while averages of the individual room temperature variations for several series are shown for the ceiling diffuser and floor register systems. In all cases, the studies were made on days with overcast skies and at times when there was no significant generation of internal heat by occupants or appliances. The mechanical differential of the thermostat was approximately 2° F. Because of the heat-anticipating device (a small resistance heater mounted in the thermostat casing which was energized during burner "on" periods) the measured thermostat differential was approximately 0.5° F. throughout all series.

The cyclical variations with the floor diffusers are shown for two outdoor air temperatures with which both burner and blower operation would normally be cyclic. Continuous blower operation with the 49° F. temperature was accomplished by setting the fan switch to the manual "on" position. With the exception of the living room, the average temperature variation between burner operations was reduced from 1.3° F. with cyclic blower operation to 0.9° F. with continuous blower operation. With either method of blower operation, the variations in the rooms were greater than at the thermostat.

The ceiling diffusers caused an increase in cyclical temperature variation with an increased distance above the floor. That is, the temperature variation near the ceiling was greater than that at the sitting level. This was attributed to the fact that the ceiling diffusers did not cause any appreciable air movement at the sitting and floor levels. At the sitting level, the cyclical variation was less than 1° F. with outdoor temperatures between 23° and 32° F. The Series K-1 air-flow rate was the highest and caused cyclic blower operation with a 32° F. outdoor air temperature. The average cyclical variation was 0.9° F. as compared with 0.4° F. and 0.2° F. during Series K-2 and K-3, respectively. The lower Series K-2 and K-3 variations were due to continuous blower operation which resulted from the lower air-flow rates, and were also due to less disturbance of the sitting level air with the lower air-flow rates.

The effect of continuous blower operation in reducing the cyclical variation is also illustrated by a comparison of studies conducted with the floor registers. For example, studies one and two were conducted with 35° F. outdoor air temperatures during Series L-1. With cyclic blower operation, the cyclical variation of room air temperature was 2.1° F. as compared with a variation of only 0.9° F. when the blower was locked on for continuous operation. Comparisons of studies five and six and studies seven and eight again show the same trend.

Cyclical variation was also reduced with the air-flow rate as is illustrated by a comparison of studies two, three, and four. As the air-flow rates were reduced from 800 to 600 to 400 c.f.m., the cyclical temperature variations were reduced from 0.9° to 0.5° to 0.3° F., respectively. A floor register produces a nonspreading vertical jet which must spread across the ceiling and down the opposite wall before it affects the occupied zone of the room. The floor register produces a stagnant zone in the lower levels of the room.[6] The height of the stagnant zone increases as the air-flow rate is reduced. The sitting level was within the stagnant zone with the lower air-flow rates, as was indicated by the traverses of the living room which were discussed in the preceding section. The decrease in cyclical temperature variation was attributed to decreased air motion at the sitting level as the air-flow rate was decreased.

Studies five through eight were conducted during Series L-4 with the fuel-input rate increased to 70,000 B.t.u.h. instead of the normal 45,660 B.t.u.h. The air-flow rate was 800 c.f.m. as it was during Series L-1. The greater fuel-input rate during Series L-4 resulted in larger cyclical room air temperature variations than when the input was matched to the calculated heat loss. For example, studies two and six were conducted with approximately the same outdoor air temperature and with continuous blower operation. The cyclical room air temperature variation during study two was 0.9° F. as compared with 1.4° F. during study six. Assuming bonnet and duct transmission efficiencies of 80 and 85%, respectively, the ratio of register capacity to calculated heat loss was 2.2 during study two and 3.1 during study six. The cyclical variation was also larger during mild weather as evidenced by a comparison of studies five and six with studies seven and eight. Studies five and six, which were conducted with a 32° F. outdoor air temperature, had variations of 2.3° F. and 1.4° F. with cyclical and continuous blower operation, respectively, while studies seven and eight, which were conducted with a 52° F. outdoor air temperature, had variations of 2.4° F. and 1.6° F. with cyclical and continuous blower operation, respectively. The ratio of register capacity to calculated heat loss was 3.1 during studies five and six and was 6.1 during studies seven and eight.

E. ROOM AIR TEMPERATURE VARIATION BETWEEN LEVELS

The effect of indoor-outdoor temperature difference, supply outlet location, and air-flow rate on the temperature variations between levels are summarized by the gradient curves in Figure 7. The gradients were obtained in turn from curves of the average temperature differences between the sitting and other levels throughout the house plotted against the 6:30 a.m. indoor-outdoor temperature difference. Therefore, the gradients represent average conditions rather than temperatures which existed under any specific set of weather conditions. For this reason, the average temperature difference from the control temperature is shown rather than actual temperatures.

With every type of outlet there is associated a stagnant zone within which the air motion is less than 15 f.p.m. and not directly affected by the supply outlet.[5, 6] Within the zone, air motion is due to natural convection. Outside the stagnant zone the air motion is greater than 15 f.p.m. and is caused by the supply air from the outlet. The boundary between the stagnant zone and the region of greater air motion is a horizontal plane, the stagnant layer.[5] During heating, the stagnant zone is between the floor and the stagnant layer. The maximum temperature gradient exists within the stagnant zone due to the fact that the air stratifies in layers of decreasing density going upward from the floor. Above the stagnant layer, the increased air motion breaks up the stratification and decreases the temperature gradient. The positions of the stagnant layers in Figure 7 were determined as the point where the change in the gradient curve occurred.

The floor diffusers located around the perimeter of the residence discharged the warm air in a spreading vertical pattern that partially blanketed the exposed walls with warm air. With a 40° F. indoor-outdoor temperature difference, according to the gradient, the stagnant layer was located at the 30-inch level. This indicates that the supply and room air were well mixed above the sitting level which resulted in a difference of less than 1° F. between the sitting and ceiling levels. The partial blanketing of the exposed wall with warm air prevented most of the natural movement of cold air down the wall to the floor level. The fact that the stagnant layer was not located at floor level indicates that cold air was falling from the exposed wall areas that were outside the blanketed areas. The gradient from the floor to the sitting level was slightly greater than 1° F. and was essentially the same for both the low (Series G-4) and high (Series G-11) air-flow rates.

An increase in the indoor-outdoor temperature difference to 70° F. raised the level of the stagnant layer with the floor diffusers to just above the sitting level in the case of the cooling air-flow rate (Series G-11) and up to the breathing level in the case of the heating air-flow rate (Series G-4). The Series G-11 air-flow rate caused mixing of supply and room air between the sitting and ceiling levels and resulted in a sitting-to-ceiling level temperature difference of approximately 1° F. The lower air-flow rate used during Series G-4 was not sufficient to promote mixing below the breathing level. Below the stagnant layer, the gradients were slightly greater with the 40° F. indoor-outdoor temperature difference. The increase in the gradient with indoor-outdoor temperature difference indicates that it is difficult to eliminate all of the natural movement of cold air down the outside wall. However, as will be shown, the diffusers in perimeter locations were more effective from this respect than were ceiling diffusers or floor registers.

Ceiling diffusers discharge the air in a horizontal pattern which spreads across the ceiling and down the sidewalls. As the air-flow rate is increased, the air travels farther down the walls. During heating it is quite difficult to push the supply air very far down the wall into the occupied zone. The result is that the stagnant layer usually occurs at a much higher level than it does with the floor diffusers as is indicated in Figure 7. The ceiling level air temperature thermocouples were in the discharge jets from the ceiling diffusers and for that reason the gradient curves do not include ceiling level air temperatures. From a study of the gradients it appears that the nearly stagnant zone extended above the breathing level and for that reason the stagnant layer is indicated as occurring above the breathing level. The gradients are shown for Series K-1, K-2, K-3, and K-2A during which the total air-flow rates were 800, 600, 400, and 793 c.f.m. During Series K-2A, 600 c.f.m. was supplied to the first story and 193 c.f.m. was supplied to the basement. The gradients were least and nearly identical during Series

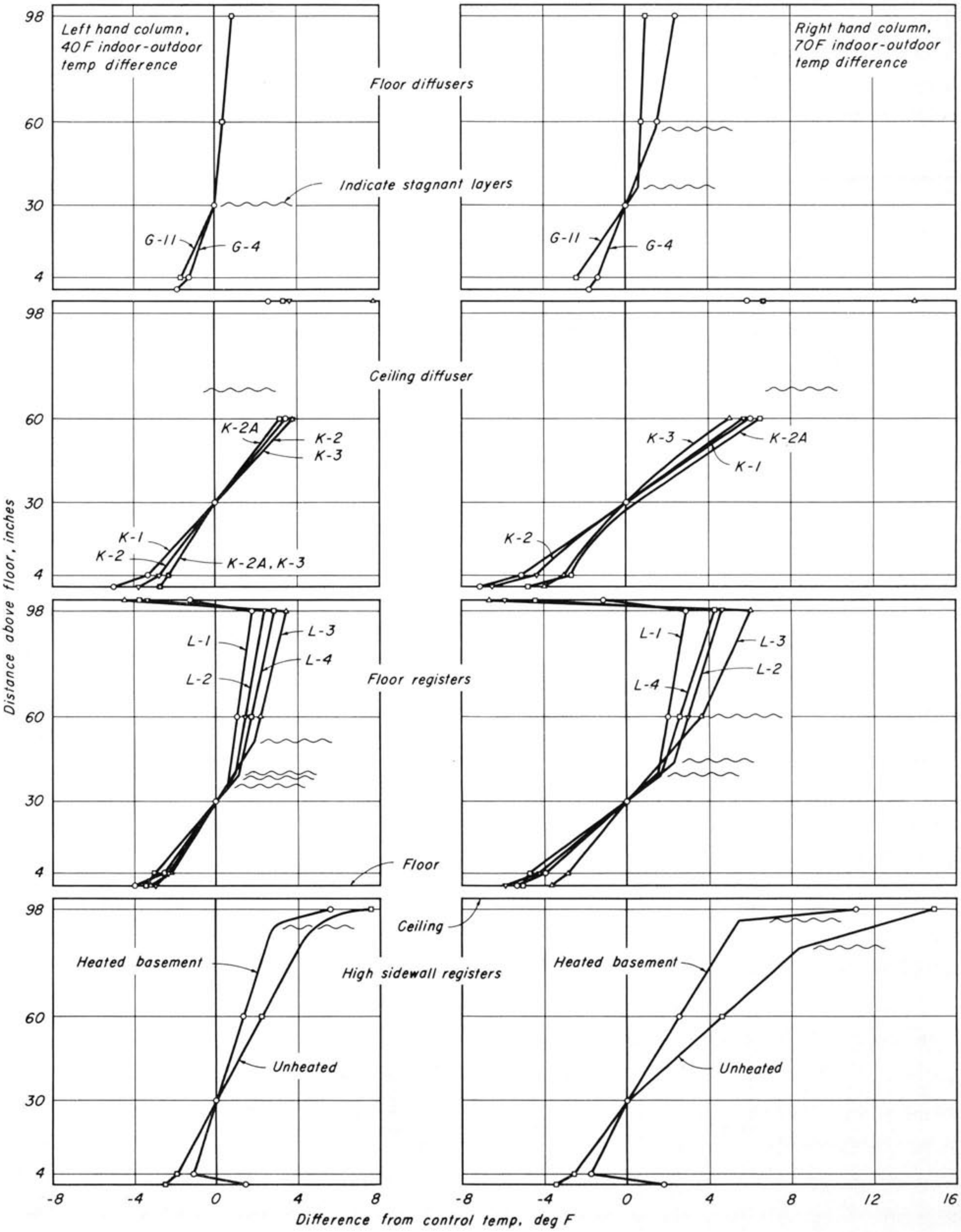


Figure 7. Gradients from average differentials

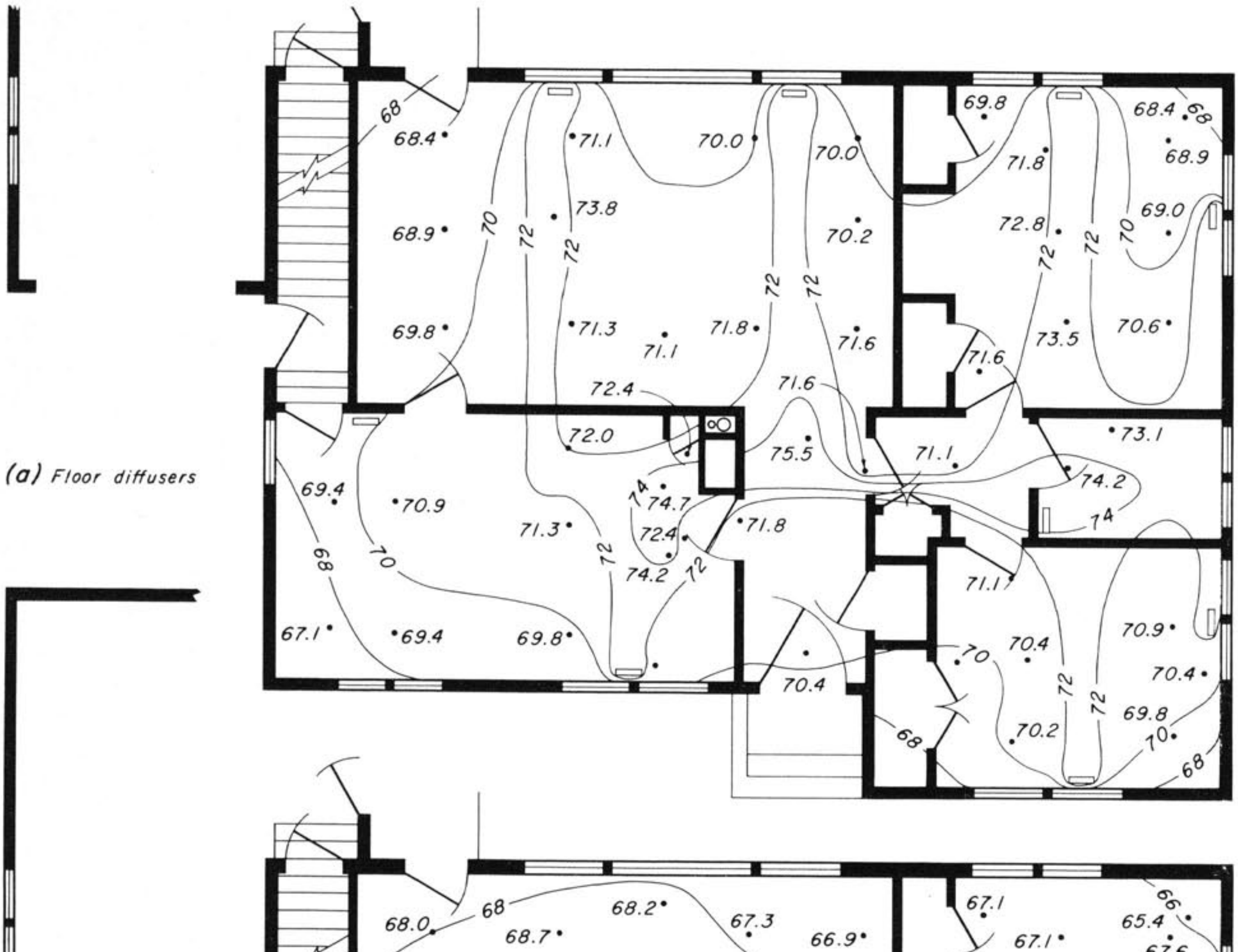
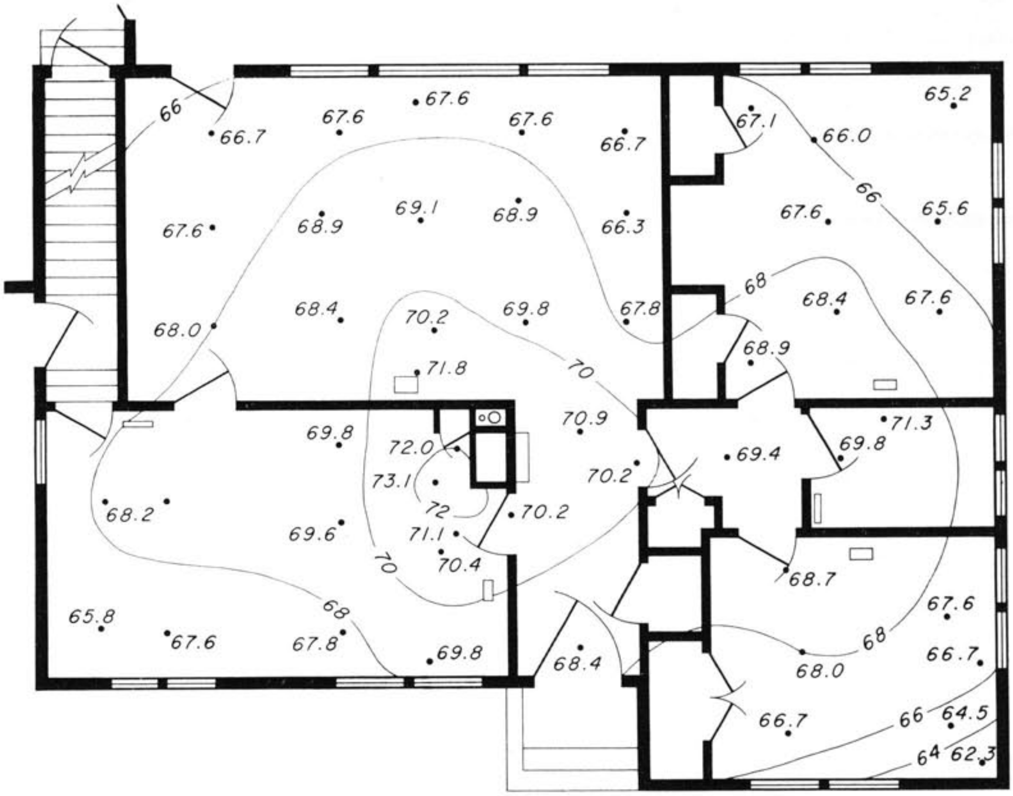


Figure 8. Floor surface isotherms



(b) Floor registers

K-3 and K-2A. In both cases this was attributed to the fact that the floor surface temperature was maintained nearer the sitting level temperature. During Series K-2A this was accomplished by circulating warm air under the first-story floor. During Series K-3, the total air-flow rate was low enough that a layer of high temperature air at the ceiling raised the ceiling surface temperatures to approximately 13.9° F. above the sitting level temperature which resulted in warming of the floor by radiant heat exchange.

Floor registers discharge the supply air in a non-spreading vertical jet which, after it reaches the ceiling, spreads across the ceiling and down the walls in a pattern similar to that of the ceiling diffusers. The air-flow rates were 800 c.f.m. during Series L-1 and L-4, 600 c.f.m. during L-2, and 400 c.f.m. during L-3. The stagnant layers occurred at lower levels than with the ceiling diffusers and were affected by the air-flow rates. The stagnant layer's height above the floor was increased as the air-flow rate decreased. Within the nearly stagnant zone the gradients were identical to those that existed with the ceiling diffusers. The Series L-3 gradient was least with a 70° F. indoor-outdoor temperature difference because of a reduction in the floor surface-to-sitting level temperature variation. In this case the increased floor surface temperature was attributed to increased heat loss from the ductwork with the low air-flow rate which raised the temperature of the air below the floor. Ceiling surface temperatures were up to 20° F. lower than with the ceiling diffuser system.

The fuel-input rate was increased from 45,660 to 70,000 B.t.u.h. during Series L-4 but the resulting gradient was essentially the same as that in Series L-2, although the air-flow rate was 800 c.f.m. as it had been during Series L-1. The difference between the L-1 and L-4 gradients above the stagnant layer was attributed to the greater temperature rise through the furnace during L-4 which increased the ceiling level air temperature.

Gradients for the high sidewall system are also shown in Figure 7. The gradients are essentially uniform from the 4-inch level to well above the breathing level. This indicates that the stagnant zone existed throughout that region. With the heated basement, the gradient was decreased due to some slight convection caused by the warmer floor. In fact, the gradients below the 30-inch level with the heated basement were as acceptable as those with the perimeter system. Above the 30-inch level the effect of the heated basement is even more apparent. One effect was to reduce the ceiling level air temperatures which is quite evident in the case of the gradients for the 70° F. indoor-outdoor temperature difference.

The gradients for all of the systems show that maintenance of a layer of warm air below the floor is beneficial to the room air temperature gradients. The best gradients existed with the perimeter system where the outside walls were partially blanketed by warm air from the floor diffusers. The perimeter system gradients were improved when the air-flow rate was low and heat loss from the branch ducts provided fairly uniform addition of heat under the floor. With the floor register and ceiling diffuser systems the introduction of heat below the floor was not as uniform and the gradients were not as good. The greatest amount of heat was introduced into the basement during the heated-basement studies conducted with the high sidewall system where a sufficient volume of warm air was introduced to satisfy the 26,285 B.t.u.h. calculated heat loss of the basement. The gradients with the high sidewall system and heated basement were not quite as good as with the perimeter system but better than those with either the inside-wall floor register system or the ceiling diffuser system.

F. FLOOR SURFACE TEMPERATURES

Floor surface temperatures were affected primarily by addition of heat under the first-story floor. With the floor outlets this was accomplished by heat losses from the duct work. In the case of the ceiling diffusers, the duct work was in the attic and heat was added below the floor by supplying warm air at the basement ceiling level (Series K-2A).

Figure 8 shows a comparison between floor surface isotherms that existed with the floor diffusers, ceiling diffusers, and floor registers. The temperatures were measured at 52 locations during early morning hours when the outdoor air temperature was approximately 32° F.

With the floor diffuser system, only the exposed corners had temperatures below 70° F. The minimum floor surface temperature of 67.1° F. occurred in one of the extreme exposed corners. The highest temperatures were measured immediately above the bonnet and ducts with a maximum of 75.5° F. and only a small area had a temperature above 74° F. The advantage of locating supply diffusers along exposed walls is that heat addition below the floor is more uniform as is evidenced by the location of the 70° F. isotherm. In areas with diffusers along the exposed wall the 70° F. isotherm was very near the wall. Along the east wall, which had no diffusers, the 70° F. isotherm extended considerably farther into the room.

The floor surface isotherms for the ceiling diffuser system show a gradual decrease in temperature from the furnace area toward the exposed walls. These temperatures were measured during Series K-3 where, as was discussed in the section on room air temperature differentials, the floor surface temperatures were maintained by radiant heat exchange with the ceiling surface. Compared on a location-by-location basis, the temperatures were approximately 2° F. lower than those with the floor diffusers.

The average floor surface temperature (average of temperatures measured at 52 locations) during Series K-1 was 67.6° F. at 40° F. indoor-outdoor temperature difference and decreased to 63.0° F. at 70° F. indoor-outdoor temperature difference. The average floor surface temperature during Series K-2A was 69.3° F. with a 40° F. indoor-outdoor temperature difference and decreased to 66.0° F. with a 70° F. indoor-outdoor temperature difference. Reduction of the air-flow rate to 400 c.f.m. during Series K-3 increased the average floor surface temperatures above the Series K-1 temperatures by 1.0° F. and 1.8° F. with 40° F. and 70° F. indoor-outdoor temperature differences, respectively.

Floor surface temperature isotherms for the floor register system show that temperatures were in between those with the other two systems. The heat losses from the basement ducts raised the floor surface temperatures above those that existed with the ceiling diffuser system. Since the floor register duct system was concentrated toward the center of the house, the temperatures near the exposed walls were less than with the floor diffusers.

The average floor surface temperature (again the average of 52 temperatures) during Series L-1 was approximately the same as during K-2A with the ceiling diffusers where heat was added under the floor. The average during Series L-1 was 69.1° F. with a 40° F. indoor-outdoor temperature difference and decreased to 65.8° F. as the indoor-outdoor

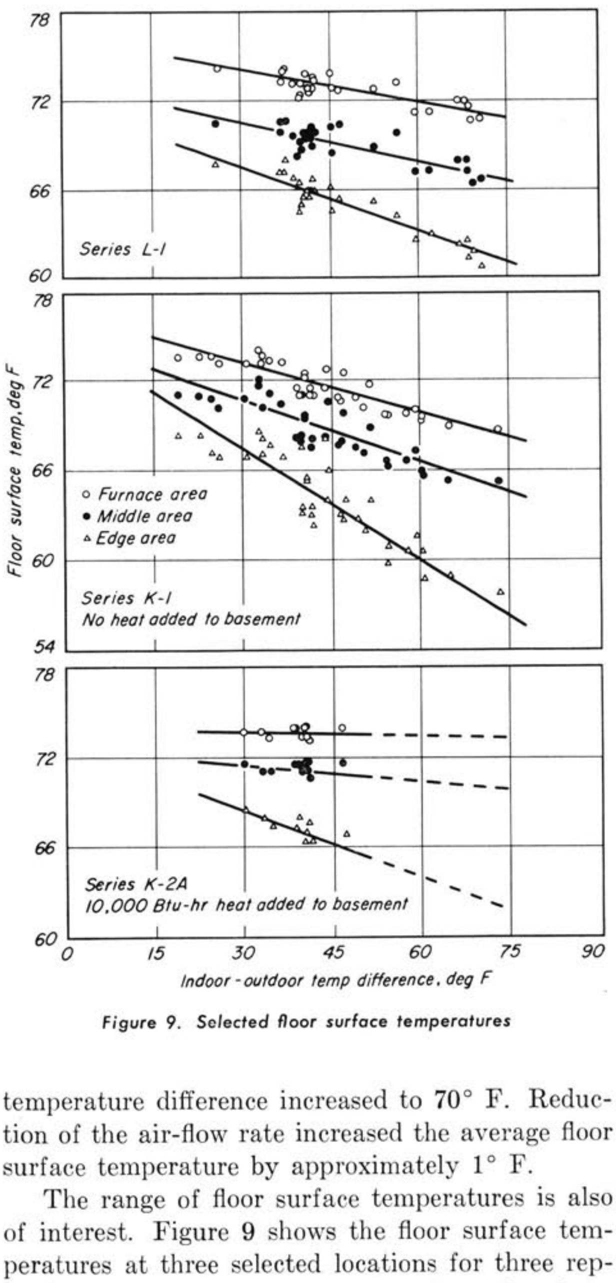


Figure 9. Selected floor surface temperatures

temperature difference increased to 70° F. Reduction of the air-flow rate increased the average floor surface temperature by approximately 1° F.

The range of floor surface temperatures is also of interest. Figure 9 shows the floor surface temperatures at three selected locations for three representative series, K-1, K-2A, and L-1. One location was directly above the furnace near the center of the residence (designated as furnace area), a second location was approximately 18 inches from the southwest corner of the residence (designated as edge area), and the third location was at an intermediate location representing the majority of the floor surface (designated as middle area). The graph for Series K-1 is representative of all series in which ceiling diffusers were utilized except K-2A, which had direct heat addition to the basement, and which is also shown in Figure 9. The third graph is for Series L-1 and is representative of all series in which floor registers were utilized as the supply outlets.

The lowest floor surface temperatures occurred when ceiling diffusers were utilized with no direct heat addition to the basement (K-1). These temperatures were considerably higher when floor registers were utilized because of the duct heat losses in the basement. The warmest floors occurred when heat was added directly to the basement during K-2A.

At 70° F. indoor-outdoor temperature difference (approximately 0° F. outdoor air temperature) the edge area temperatures were 57.2° F. during K-1, 61.8° F. during L-1, and 62.5° F. (extrapolated) during K-2A; the furnace area temperatures at 70° F. indoor-outdoor temperature difference were 68.6° F. during K-1, 71.0° F. during L-1, and 73.4° F. (extrapolated) during K-2A; and the middle area floor surface temperatures at 70° F. indoor-outdoor temperature difference were 65.1° F. during K-1, 66.2° F. during L-1, and 72.7° F. during K-2A.

During a heating investigation with floor diffusers[4] when no heat was added directly to the basement, the edge area floor surface temperature at 70° F. indoor-outdoor temperature difference was 65.5° F., 3.0° F. higher than the temperature at the same location during Series K-2A. This was partly due to lower floor-level air temperatures and partly due to lower basement air temperatures with the ceiling diffuser system.

G. WINTER AIR CONDITIONER OPERATING CHARACTERISTICS

The curves shown in Figure 10 are representative of those which resulted with the lowest air-flow rate (295 c.f.m.) and the highest air-flow rate (800 c.f.m.). Flue gas temperature is an indication of the energy lost in the flue gases. The higher the temperature, the greater the loss. Blower operation time and number of operations are dependent on the air-flow rate and the fan switch setting. In all series, the bonnet switch was set to stop the blower when the bonnet temperature dropped to 80° F. Therefore, the curves show the effect of air-flow rate. Hours of burner operation is the product of the number of cycles per day and the cycle length. As the indoor-outdoor temperature difference increases, the cycle length also increases until continuous burner operation is approached at design

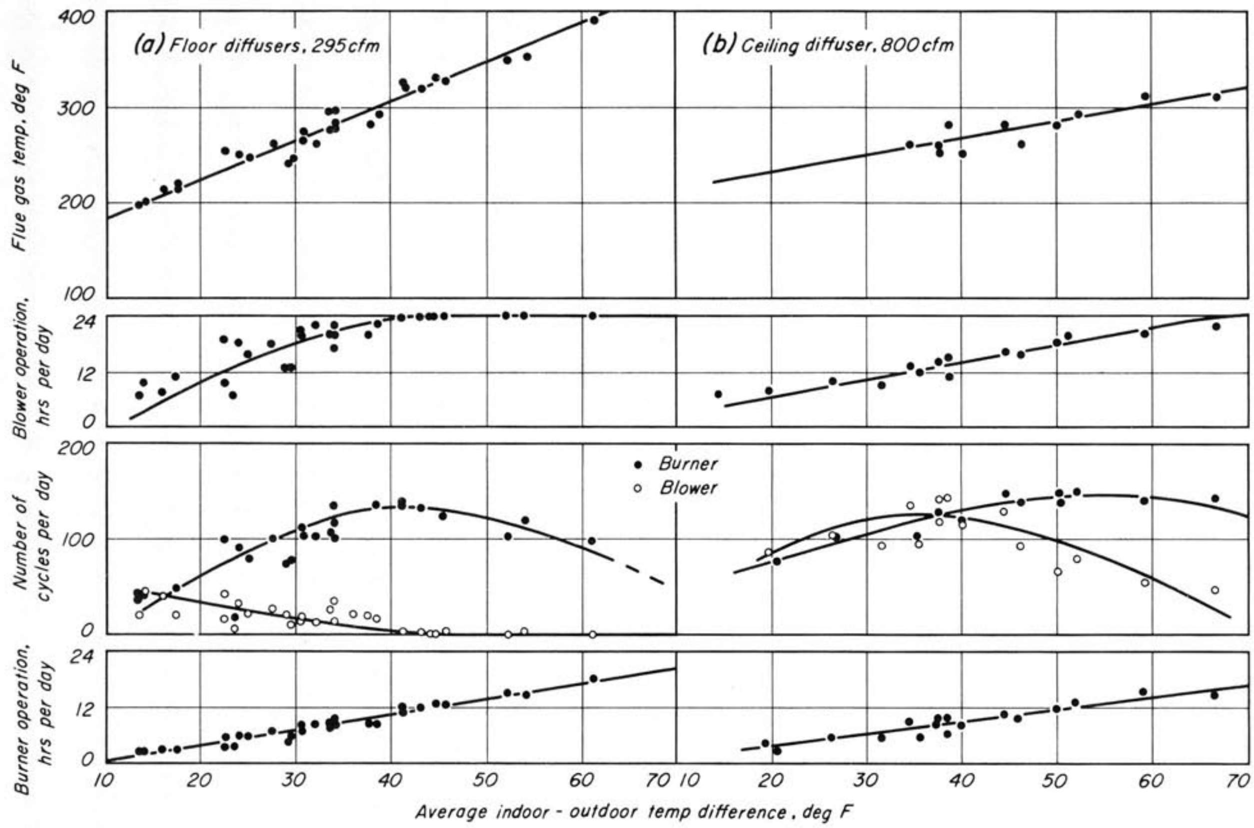


Figure 10. Furnace operating characteristics

conditions. The number of cycles per day reaches a maximum and then decreases as the burner operation becomes more nearly continuous.

Since the residence was occupied during all of the heating investigations, a part of the energy required for heating the house was supplied by the electrical energy supplied to the blower motor, the household appliances, and the lighting. The furnace was oversize for the house and fuel input was reduced to match the calculated heat loss of the residence. However, the secondary air was not always adjusted to maintain the maximum combustion efficiency. The bonnet efficiency was affected by the air-flow rate and with low air-flow rates (as little as one-half the 100° F. rise air-flow rate for the full rated fuel input) the bonnet efficiency dropped to as low as 58%. This would not occur in the usual field installation since fuel input is normally not reduced below 80% of the rate listed on the A.G.A. approval label. For those reasons the systems are compared on the basis of total energy input to the house rather than on the basis of fuel consumption. Curves of total energy input as a function of indoor-outdoor temperature difference were drawn for all of the series with floor diffusers, floor registers, and ceiling diffusers. Values from the performance curves for 40° F. and 70° F. indoor-outdoor temperature differences have been listed in Table 5. Total energy requirements are affected by a number of variables such as wind and solar intensity. No attempt was made to correlate all of these variables, and the values listed in Table 5 are average values of total energy input to the residence. In addition, Table 5 includes the results of some special studies where the burner was operated continuously to permit measurement of combustion and bonnet efficiencies and also the air temperature rise through the furnace. The combustion efficiency decreased with the fuel-input rate. The decrease was attributed to a less favorable fuel–air ratio. The rated input of the furnace was 70,900 B.t.u.h. and as the fuel input was reduced the secondary air was not adjusted accordingly. This resulted in excess secondary air which was heated and then escaped up the flue. The excess secondary air was also indicated by the reduced CO_2 content of the flue gases at the lower fuel-input rates. Maximum combustion efficiency occurred

Table 5

Summary of Furnace Operating Characteristics

Series	Floor Diffusers		Floor Registers				Ceiling Diffusers			
	G-4	G-1	L-1	L-2	L-3	L-4	K-1	K-2	K-2A	K-3
Air-Flow Rate, Standard c.f.m.[a]	295	530	798	615		774	757		793*	404
Furnace Firing Rate, B.t.u.h.	45,660	45,660	46,660	45,800		70,900	45,800	45,660	60,360	45,800
Combustion Efficiency, %[a]	72.0	76.0	75	74		78	76			72
Bonnet Efficiency, %[a]	58.0	70.5	71.5	72.0		71.0	68			62.8
Temperature Rise in Furnace, °F.[a]	82.8	56.0	39.0	49.8		60.3	38.6			66.0
CO_2 Content of Flue Gases, %[a]	...	...	4.9	5.0		8.3	4.9			4.8
Total Energy Input,[b] B.t.u.h. per Day $\times 10^{-3}$										
Indoor-Outdoor Temperature Difference = 40° F.	630	585	630	620	600	...	610	610	710	...
Indoor-Outdoor Temperature Difference = 70° F.	1110	1040	1020	1015	...	...	975	...	...	...
Number of Burner Operations per day										
Indoor-Outdoor Temperature Difference = 40° F.	135	120	146	160	159	126	129	157	148	...
Indoor-Outdoor Temperature Difference = 70° F.	50	78	118	85	...	...	123	...	...	...
Maximum Number of Operations per Day	135	125	153	167	162	118	148	160	152	157
Indoor-Outdoor Temperature Difference at Which Maximum Occurred, °F.	40.0	44.5	49.5	48.0	35.5	38.0	52.0	39.0	43.5	43.0
Average Daily Flue Gas Temperature, °F.										
Indoor-Outdoor Temperature Difference = 40° F.	305	255	260	360	320	270	270	280	290	...
Indoor-Outdoor Temperature Difference = 70° F.	430	300	360	420	...	...	320	...	...	...

* Approximately 600 c.f.m. supplied to first story, balance to basement.
[a] Measured with burner operating continuously.
[b] Fuel with a heating value of 1000 B.t.u.h. per cu. ft.

with a fuel-input rate of 70,900 B.t.u.h. which was accompanied by a flue gas CO_2 content of 8.3%.

Bonnet efficiency increased from 58% with the low air-flow rate of 295 c.f.m. to between 71% and 72% with the high air-flow rate. This was attributed to improved heat transfer between the heat exchanger and the warm air as the velocity on the air side of the heat exchanger was increased.

The balance of the table is based on average indoor-outdoor temperature differences as recorded on the automatic temperature recorders. During those series where the basement was not heated directly the total energy requirements were similar regardless of the location of the supply outlets or the efficiency of the furnace. Heat losses from the basement duct work were not lost as far as total house heating was concerned. The losses helped to improve the first-story floor surface temperatures. In addition, all of the heat carried with the flue gases was not lost since the chimney was located in the center of the house and some of the heat was regained to the first-story rooms. Instrumentation was not included to measure the heat regain, but undoubtedly this also helped to equalize the total energy requirements of all the systems. The fact that the total energy inputs during Series K-1 and K-2 were comparable to those when floor diffusers and floor registers were used shows that the insulation of the attic duct work was sufficient to prevent excessive heat losses to the attic. During Series K-2A warm air was supplied directly to the basement which increased the total energy input above that required for the other series.

Table 6 is a summary of blower operating characteristics and duct pressure loss characteristics. With the floor register and ceiling diffuser systems, the bonnet static pressure, which is a measure of the pressure loss of the supply duct system, varied as the square of the volume of air flowing through the duct system, except for Series K-2A. During Series K-2A, the total air-flow rate was approximately the same as during Series K-1. However, 200 c.f.m. was introduced into the basement through stub ducts. The remaining 600 c.f.m. was distributed through the attic plenum and duct system, and the attic bonnet pressure was approximately the same as for Series K-2 where the total air-flow rate was 600 c.f.m. The floor diffuser supply duct system damper

Table 6

Summary of Blower Operating Characteristics and Pressure Losses in Duct System

Series	Floor Diffusers		Floor Registers				Ceiling Diffusers			
	G-4	G-1	L-1	L-4	L-2	L-3	K-1	K-2A	K-2	K-3
Air-Flow Rate, c.f.m. Standard Air	295	530	806	794	600	393	798	781	600	408
Bonnet Static Pressure, Inches of Water	0.106	0.275	0.116	0.111	0.064	0.026	0.211	0.107	0.116	0.051
Return Static Pressure, Inches of Water	−0.020	−0.053	−0.103	−0.099	−0.057	−0.025	−0.100	−0.100	−0.055	−0.027
Attic Bonnet Static Pressure, Inches of Water	...	...	...	...	...	...	0.154	0.083	0.087	0.043
Total External Static Pressure Loss, Inches of Water	0.126	0.328	0.219	0.210	0.121	0.051	0.311	0.207	0.171	0.078
Blower Power Requirement, Kw.	0.13	0.24	0.29	0.288	0.17	0.12	0.31	0.28	0.19	0.13
Hours of Blower Operation per Day Indoor-Outdoor Temperature Difference = 40 °F.	24	20	16	15	22	24	14	18	22	24
Number of Blower Operations per Day Indoor-Outdoor Temperature Difference = 40 °F.	3	34	100	130	20	1	120	115	60	15
Indoor-Outdoor Temperature Difference Above Which Blower Operated Continuously, °F.	40	60	70	60	50	35	70	50	50	40

adjustments were not similar during Series G-1 and G-4. For that reason the pressure losses did not vary as the square of the air-flow rate.

The same return air system was used for all series and had originally been designed for a static pressure loss of 0.05 inch of water with an air-flow rate of 800 c.f.m. The floor register and floor diffuser systems were designed for air-flow rates of 800 c.f.m. with bonnet static pressures of 0.15 inch of water. With the design air-flow rate, the furnace bonnet static pressure was less than the 0.15 inch of water with the floor register system and the attic bonnet static pressure was only 0.154 inch of water with the ceiling diffuser system. The return air system losses were approximately 0.100 inch of water or approximately twice the design value of 0.05 inch of water. In the case of the floor register system, the supply system design losses were conservative and compensated for the excess return system loss which resulted in a total external static pressure loss from 0.01 to 0.02 inch of water in excess of the design value of 0.20 inch of water. The loss of the attic part of the ceiling diffuser supply system was approximately equal to the design value of 0.15 inch of water, but with the additional losses of the connecting duct between the furnace and attic bonnet and that of the return system, the over-all external static pressure loss was 0.311 inch of water.

The blower was set to cut off at approximately 80° F. bonnet air temperature and this setting resulted in almost continuous blower operation with an indoor-outdoor temperature difference of 40° F. or more and with the lower air-flow rates (Series G-4, L-3, K-3). This type of blower operation was desirable because it resulted in nearly continuous heat addition to the occupied space (though not at a constant rate) and the cyclic room air temperature variations were less, as was discussed earlier. Nearly continuous heat input is the important feature of the Comfort Air Circulation principle. Continuous blower operation could be accomplished with any air-flow rate merely by locking the blower switch on. However, with the C–A–C air-flow rate, continuous blower operation is accompanied by circulation of warm air above room temperature which prevents discharge of cool air from the supply outlets.

H. INDOOR RELATIVE HUMIDITY

The average daily indoor relative humidity

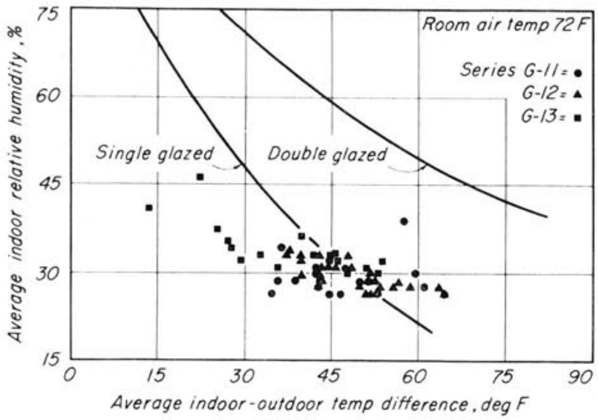


Figure 11. Average indoor-outdoor temperature difference

measured during the 1955–56 heating season is represented as a function of average indoor-outdoor temperature difference in Figure 11. Because of the large scatter of points no attempt was made to construct a representative curve. The variations were caused by variations in outdoor air humidity for specific indoor-outdoor temperature differences and variations in the living habits of the occupants, i.e., variations in the internal moisture release from washing, bathing, cooking, etc. The two curves in Figure 11 represent the calculated indoor relative humidity at which condensation will occur on the inside surfaces of glass panes at specific indoor-outdoor temperature differences. These curves were obtained by assuming a room air temperature of 72° F. and calculating the temperature drop across the inside film, based on *Heating, Ventilating, and Air Conditioning Guide*[11] data. The temperature drop was subtracted from the room air temperature to yield the inside glass surface temperature, which determined the relative humidity at which condensation will occur. If, for a specific indoor-outdoor temperature difference, the indoor air relative humidity is above the single-glazed curve, condensation would be expected to occur on a single-glazed window. If the relative humidity is between the single-glazed and double-glazed curves, double-glazed windows should be expected to prevent condensation. All first-story windows in the residence were equipped with storm sash, and condensation occurred on rare occasions at a source of moisture, for example, above the kitchen sink during dish-washing periods; but upon elimination of the moisture source the condensation on the windows was re-evaporated by the room air.

IV. PERFORMANCE OF SYSTEMS DURING COOLING

A. EXPERIMENTAL CONDITIONS

The conditioned space consisted of all first-story rooms. The basement was cooled only by heat gain to the ducts and cooling unit and by air leakage between the first story and the basement. The thermostat, whose location in the entrance hall was the same as during the heating season investigations, was adjusted to maintain a temperature of approximately 75° F. at the 30-inch level.

The three systems investigated were the floor diffuser, ceiling diffuser, and floor register systems described previously. The floor diffuser system was under investigation during the 1954 and 1955 cooling seasons, during which six test series were conducted. The four test series conducted in 1955 were concerned with temperature balance, air-cooled condensing, and night-air cooling, each of which is discussed separately in later sections. The two 1954 cooling season series, designated as Series S54-1 and S54-2, were to determine comfort conditions produced by floor diffusers during cooling and to determine the performance characteristics of a small-pipe air distribution system. During both S54-1 and S54-2 the air-flow rate was set at 560 c.f.m. (300 c.f.m. per nominal ton of cooling unit capacity) and the only difference between test series was in the operation of the blower. During S54-1 the blower was operated continuously and during S54-2 the blower was cycled with the compressor.

Only one test series (S57-1) was conducted with the ceiling diffusers during cooling. The air-flow rate was 800 c.f.m., and the blowers were operated continuously.

Two series were conducted with the floor registers. During the first series, S57-2, the air-flow rate was 800 c.f.m. and during the second series, S57-3, the air-flow rate was reduced to 600 c.f.m. The blower was operated continuously during both series.

B. SUPPLY AIR CHARACTERISTICS

The supply air from the floor diffusers was projected vertically and spread in a fan-shaped pattern in a plane parallel to the wall. The vane settings of the diffusers determined the spread of the supply air. During cooling, it is important to project the cool supply air to near the ceiling level and, therefore, too great a spread is not desirable. The supply jet will fold back on itself unless it is projected upward to the ceiling. If projected to the ceiling level, the supply air will then spread across the ceiling and drop into the room after mixing with the ceiling level air.[5]

The supply air from the ceiling diffusers was projected horizontally and remained near the ceiling until its velocity decreased to less than 100 f.p.m., although the air was not directed equally in all directions. For example, 2 feet from the center of the living room east diffuser, and near the ceiling, the supply air velocities in the east and south directions were 200 f.p.m., while at the same distances north and west the velocities were 120 f.p.m. The branch supply duct approached the diffuser from the northwest and the non-uniformity of air flow was attributed to the effect of the elbow. The supply air velocities 4 feet from the center of the living room east diffuser were 70, 60, 85, and 100 f.p.m. in the west, north, east, and south directions, respectively. The distances of 4 feet were near the walls and near the center of the living room.

Supply air characteristics of two representative floor registers during isothermal operation (supply air temperature equal to room air temperature) are shown in Figure 12. Multiples of 100 f.p.m. isovels (lines connecting points of equal velocity) are indicated. The velocities were measured in a plane parallel to the adjacent wall and at the center of the register. The isovels indicate that the supply air was projected upward with little spread and that the height of a given isovel was dependent upon the supply velocity, and increased with the supply velocity. The supply air from floor registers strikes and spreads partially over the ceiling and drops into the room after mixing with the room air at the ceiling level[5] in a pattern similar to that of the floor diffusers. For a given supply velocity the floor register will spread the air over more of

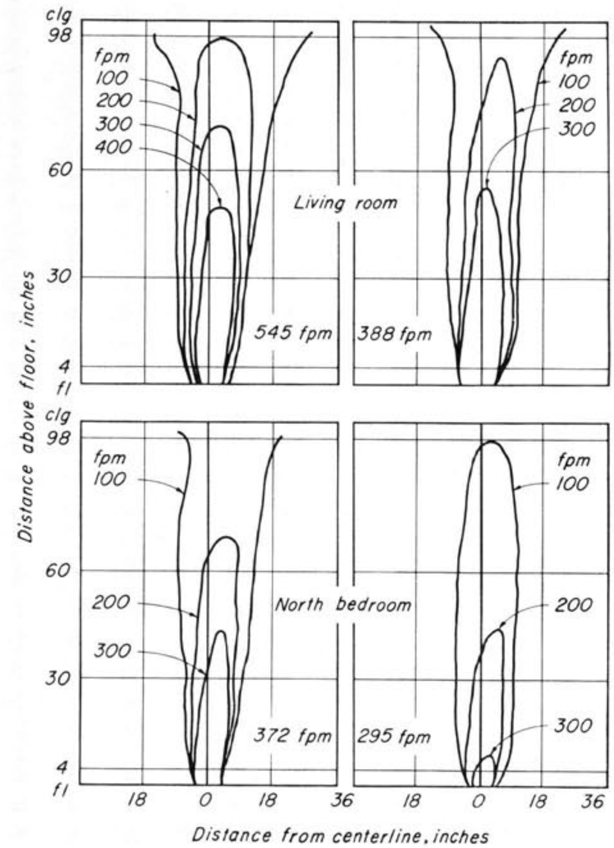


Figure 12. Isothermal supply air characteristics of two floor registers

the ceiling area than will the floor diffuser. This is due to the fact that the floor register confines the supply air in a nonspreading vertical jet until it strikes the ceiling.

C. UNIFORMITY OF ROOM AIR TEMPERATURES AT VARIOUS LEVELS

A special study with the perimeter floor diffusers was conducted during the 1955 cooling season to determine the uniformity of temperature at four levels in the living room. Temperatures were measured at six locations in addition to the locations of the two permanent thermocouple standards. On this day, the outdoor air temperature was 90° F. and the compressor was operating continuously. The maximum variations at the floor, sitting, breathing, and ceiling levels were 0.9° F., 1.0° F., 1.4° F., and 0.5° F., respectively. This indicated that the supply air was well mixed with the room air and there were no localized areas of severely cool or warm air.

With the ceiling diffusers, temperatures were measured at 50 locations in each of the planes of the sitting and floor levels with portable thermocouples. The temperatures were measured with the compressor operating and with an outdoor air temperature of approximately 87° F. At each level, the temperatures varied approximately 1° F., with the warmer air near the exposed wall. The small variation in temperatures indicates that the cool supply air from the ceiling outlets had been thoroughly mixed with the room air.

Temperature measurements were also made at the 50 locations in the planes of the sitting and floor levels to determine uniformity of temperatures with the inside-wall floor registers. The temperature measurements were also made with the compressor operating continuously and with an outdoor air temperature of approximately 87° F. At the higher air-flow rate (Series S57-2, 800 c.f.m.) the variation in temperature at both the sitting and floor levels was approximately 2° F. The lower temperatures occurred near the register and the inside wall, and the higher temperatures occurred near the exposed wall. With the lower air-flow rate (Series S57-3, 600 c.f.m.) the temperature variation at the sitting level was 2° F. and was 3° F. at the floor level. This was due to the fact that it was not possible to project the cool supply air to the ceiling level with the lower air-flow rate and, therefore, some of the cool air dropped to the floor level, which created a low temperature area.

Adequate mixing of supply and room air with the floor diffusers and floor registers is dependent on spreading the cool supply air over the ceiling. The supply velocity must be at least 500 f.p.m. to accomplish this. The ceiling outlets supply air at the proper level in the room for mixing, and good distribution is dependent on the spreading of the supply air over the ceiling by the diffuser.

D. ROOM AIR TEMPERATURE VARIATION BETWEEN LEVELS

Figure 13 is a summary of the average gradients which existed in the living room with each type of supply outlet. The gradients were obtained from curves of the average temperature difference between the sitting and other levels which existed at 3:00 p.m. with an indoor-outdoor temperature difference of 20° F. Under these conditions the compressor was always in operation. Therefore, the gradients represent the performance characteristics of each type of outlet under maximum conditions.

Table 7

Room Air Temperature Variations in Living Zone (4-inch to 60-inch Level) at 20° F. Indoor-Outdoor Temperature Difference

Series	Outlet Type		Living Room E	Living Room W	South Bedroom S	South Bedroom W	Bath	North Bedroom N	North Bedroom W	Dinette	Kitchen
S54-1 and S54-2	Perimeter Floor Diffusers	Average Supply Velocity, f.p.m.	485	475	189	283	110	157	330	398	503
		Temperature Variation, °F.	2.3		4.7		7.2	5.6		2.4	2.0
S57-2	Inside Wall	Average Supply Velocity, f.p.m.	537		429		90	367		648	706
		Temperature Variation, °F.	1.1		1.3		3.9	1.7		0.6	1.0
S57-3	Floor Registers	Average Supply Velocity, f.p.m.	376		334		72	298		459	557
		Temperature Variation, °F.	4.2		2.0		3.8	3.0		2.4	0.7
S57-1	Ceiling Diffusers	Air-Flow Rate, c.f.m.	147	181	135		38	131		52	127
		Temperature Variation, °F.			0.6		−0.2	0.2		−0.1	0.2

Each outlet has associated with it a definite pattern in which the cool supply air is discharged. The pattern determines the general room air motion and, except for the ceiling diffuser, each supply outlet will create a zone between the ceiling and a stagnant layer within which the air will remain relatively stagnant.

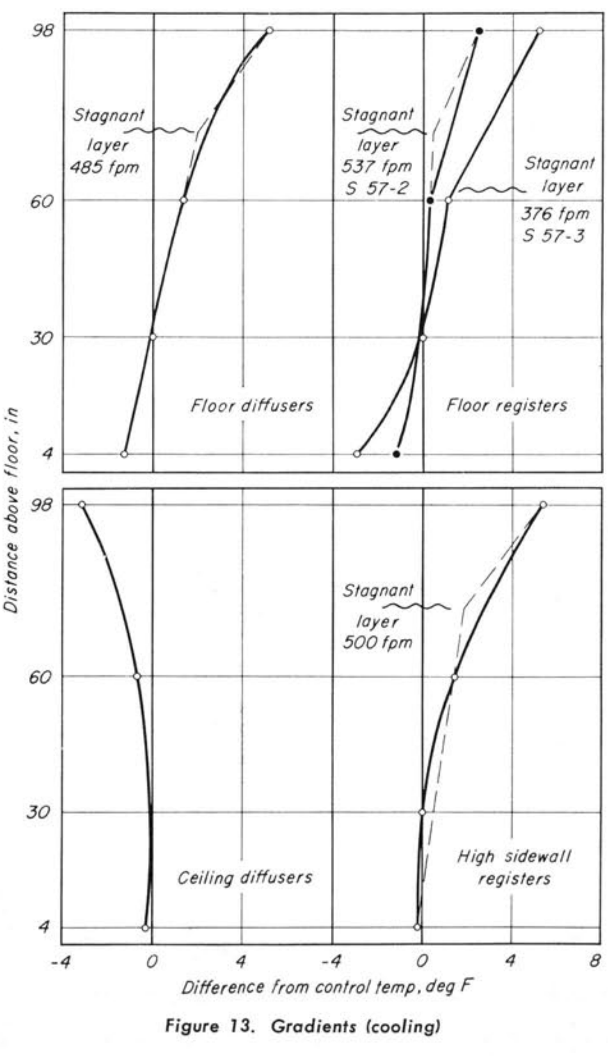


Figure 13. Gradients (cooling)

The floor diffuser with its vertical spreading discharge pattern must have a sufficient supply velocity to project the primary air up to the ceiling where it drops into the room. This provides mixing of the primary and room air. The mixture, or total air, then descends into the occupied zone of the room.[5] If the supply velocity is insufficient to project the primary air to the ceiling, the supply jet will fold back on itself and poor mixing of primary and room air will result. The total air then falls and causes low temperatures at the floor level. In Figure 13 the change in gradient above the 60-inch level indicates the existence of a stagnant layer between the 60-inch level and the ceiling. The thermocouple standards were located approximately 6½ feet from the outside wall. The existence of the stagnant zone at the thermocouple standard indicates that the primary air dropped into the room before it reached the standard and left the upper region of the room relatively undisturbed. The intersection of the dashed lines indicates the approximate location of the stagnant layer. With a supply velocity less than 485 f.p.m. the stagnant layer would move down. In University of Illinois Engineering Experiment Station Bulletin No. 442[6] it is recommended that the minimum supply velocity should be 500 f.p.m. for cooling with floor diffusers, with a 15° F. room air to supply air temperature difference. At a 20° F. temperature difference it is recommended that the minimum supply velocity should be 750 f.p.m. During the 1954 season the room air to supply air temperature difference was approximately 15° F. As shown in Table 7, in the room in which the supply velocity was near 500 f.p.m. — living room, dinette, and kitchen — the 4-inch to 60-inch level differentials were 2.3° F., 2.4° F., and 2.0° F., respectively. In the other three rooms, the differentials were 4.7° F., 7.2° F., and 5.6° F., and the maximum supply velocity for these rooms was 330 f.p.m.

Floor registers discharge the primary air in a nonspreading vertical jet, as was described in the section on supply air characteristics. As with the floor diffusers, the primary air must reach the ceiling to properly mix with the room air. Figure 13 shows two gradients for floor registers. With a 537 f.p.m. supply velocity, the primary air was mixed with the room air to reduce the floor-to-breathing level (4-inch to 60-inch) differential to 1.5° F., and the stagnant layer was apparently above the 6-foot level. A special study during S57-2 located the drop zone between the permanent thermocouple standards and the outside wall. The primary air apparently passed between the 60-inch and 98-inch levels without disturbing the ceiling level air at the standards. Reduction of the supply velocity to 376 f.p.m. increased the floor-to-breathing level differential to 3.9° F., and most of the differential was between the floor and sitting levels. This shows that the primary air did not reach the ceiling and the supply jet folded back on itself and descended to the floor level where it depressed the floor level temperature. (A special study located the drop zone during Series S57-3 between the floor register and the thermocouple standards.) The insufficient supply velocity also caused the stagnant layer to move down to near the 60-inch level.

In Bulletin No. 442[6] it is also recommended that the minimum supply velocity for cooling with floor registers should be 500 f.p.m. for a 20° F. room air to supply air temperature difference. During the cooling investigation in which the floor registers were used, the room air to supply air temperature difference was approximately 19° F. during continuous compressor operation. As is shown in Table 7, the differentials that occurred when the supply velocity was 500 f.p.m. or greater ranged from 0.6° F. to 2.4° F. In general, the differentials were greater with the lower air-flow rate used during Series S57-3. For the same supply velocities the floor registers with their nonspreading jet characteristics produced lesser differentials than did the floor diffusers.

A special study was conducted during S57-2 to determine whether the differentials measured at the locations of the two permanent standards in the living room were representative of the differentials at other locations, particularly near the outside walls. Portable thermocouple standards were positioned near the center and 1 foot from the east and south walls. The differentials measured at these locations were within 1° F. of the differentials at the locations of the permanent thermocouple standards.

During S57-3 when the compressor had operated an hour or more the ceiling-level temperatures measured in the kitchen at the permanent thermocouple standard were about 3° F. higher than the sitting-level temperatures. When the compressor had operated 30 minutes or less the ceiling-level temperatures were as much as 1.8° F. lower than the sitting-level temperatures. Since the supply velocity was constant, this indicates that the total air — the mixture of primary (supply) air and room air under the influence of the initial outlet conditions — spread farther along the ceiling at higher supply temperatures (near the beginning of the compressor operation period) than it did after the minimum supply air temperature was reached. Thus, the drop zone, the region in which the total air enters the occupied zone, was not stable but moved from near the outside wall toward the inside wall as the supply air temperature decreased. At higher supply velocities the effect of supply air temperature on the throw and drop zones was less.

Since the ceiling diffusers introduced the primary air at the ceiling level and spread it over the ceiling to the walls, the primary air opposed the natural movement of room air up the walls. This resulted in good mixing of the primary and room air to produce a differential of only 0.3° F. between the floor and breathing levels with an indoor-outdoor temperature difference of 20° F. The gradient in Figure 13 shows a decrease in temperature above the 60-inch level which was due to the layer of cool air at the ceiling level. There could be no stagnant layer with the inverse temperature gradient since the denser cool air would always move downward.

Cooling with high sidewall registers had been studied during both the 1952 and 1953 cooling seasons and performance of the system was discussed in an ASH&VE paper.[12] The living room outlet was mounted 78 inches above the floor on the north inside wall and discharged the primary air in two jets each 22 degrees from the centerline of the register. The center of each jet issuing from the register was located both when the compressor was operating (supply air temperatures between 61.5° and 64.0° F.) and when the compressor was not operating (supply air temperature approximately equal to room air temperature). In each case the average air velocity at the register face was approximately 500 f.p.m.

When the supply air was at room air temperature, the center of the jet was about 5 feet above the floor at a distance of 12 feet from the inside wall (1 foot 4 inches from the exterior wall). When the compressor was operating and the supply air temperature was lower than room air temperature, however, the center of the jet was only 2 feet above the floor at the same location (or approximately 5 feet above the floor at the standard). The gradient shown in Figure 13 existed while the compressor was operating, which meant that the jet was approximately 5 feet above the floor at the standard, and this produced a stagnant layer at approximately the 6-foot level as indicated by the intersection of the dashed lines in Figure 13.

The stagnant layer and stagnant zone above it could have been eliminated by directing the supply jets upward to spread across the ceiling which would have resulted in a gradient similar to that with the ceiling diffuser.

E. PERFORMANCE OF SUMMER AIR CONDITIONER AND DUCT SYSTEM

Cooling unit operation may be expressed[12] as a function of maximum daily dry-bulb temperature, mean daily temperature, or degree-hours above some datum temperature. Although maximum daily dry-bulb is the simplest correlation of the three it is the least reliable since it does not take into account temperatures preceding the maximum. The degree-hour index is the most difficult correlation because of the difficulty of obtaining the degree-hour information. The mean daily temperature (M.D.T.) correlation is more reliable than the maximum daily temperature index and information on M.D.T. is readily available.

The compressor operation as a function of M.D.T. is shown in Figure 14. The three plots show operation for three different seasons to illustrate the effects of two methods of blower operation, the effect of ventilation, and the effect of occupants.

The daily compressor operation time for the two test series conducted during the 1954 cooling season is shown in the upper part of Figure 14. During 8 days of the cyclic blower series the residence was unoccupied and this period is also indicated. The nominal cooling unit capacity was 19,400 B.t.u.h. More hours of compressor operation resulted when the blower was operated continuously than when the blower was cycled with the compressor. Part of the increased operation time was due to the sensible heat addition in the form of electrical energy (0.40 kw.) to the blower motor. For 12 hours of compressor operation, the blower when operated continuously would run 12 hours longer than when the blower was cycled. This additional 12 hours of blower operation added 16,400 B.t.u.h. which was equivalent to 0.85 hour of compressor operation, or approximately one-half of the difference in operation time. With 4 hours of compressor operation time the additional 20 hours of blower operation was equivalent to 1.41 hours of compressor operation, approximately the difference in compressor operation time between the two methods of blower

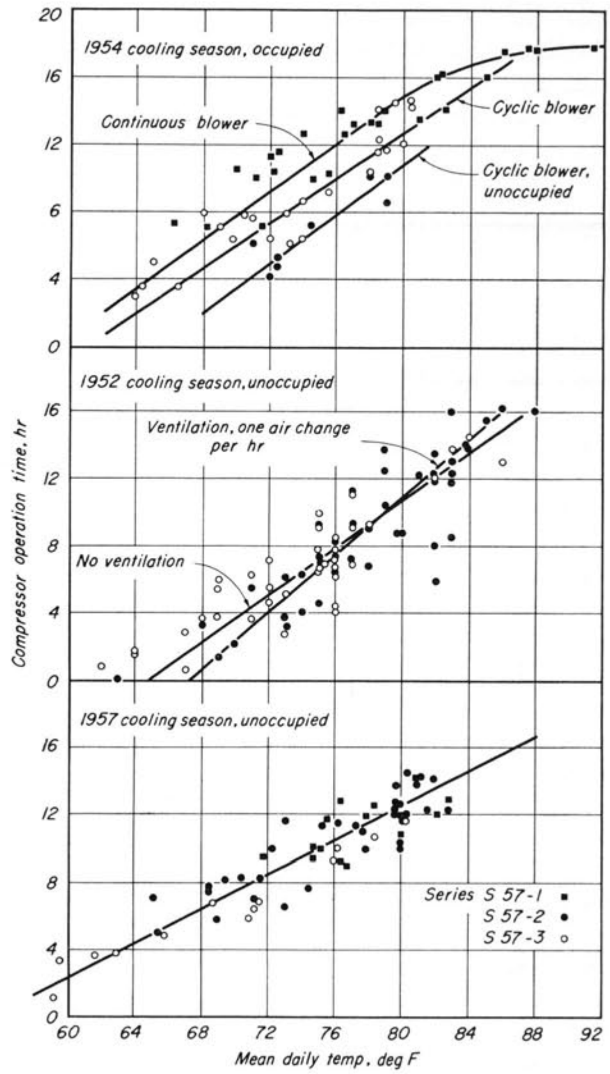


Figure 14. Effect of mean daily temperature on compressor operation

operation. For 16 hours of compressor operation the additional 8 hours of blower operation was equivalent to 0.56 hour of compressor operation, approximately one-fourth of the difference in operation time. Differences in solar radiation accounted for some difference in operation time between the two methods of blower operation. In general, less solar radiation was received during the cyclic blower operation test series than during the continuous blower operation test series. For example, during the continuous blower operation test series the total solar radiation was more than 2,000 B.t.u. per square foot per day (on a horizontal plane) on 20 of 26 test days. During the cyclic blower test series this amount of solar radiation was received on only 4 of 24 test days.

The effect of occupancy and the accompanying various heat release processes is evidenced by the decreased compressor operation time that resulted when the residence was unoccupied.

The deviation of the compressor operation time curve from a straight line resulted from an increase in indoor air temperature at higher mean daily temperatures. The maximum point shown corresponds to a day which had a maximum temperature of 110° F. and a mean daily temperature of 91° F. The indoor air temperature on that day was 81° F. at 2:30 p.m.

The datum temperatures (mean daily temperature above which compressor operation occurred) were 59° F. and 61° F. for continuous and cyclic blower operation, respectively.

For comparative purposes, the compressor operation time curve obtained during the 1952 cooling season is shown in the middle part of Figure 14. During this season the residence was unoccupied. One test series was conducted with no ventilation air and the other was conducted with ventilation air mechanically introduced into the return air duct at the rate of one air change per hour. The datum temperature obtained with no ventilation air was 65° F., approximately the same as during the 1954 cooling season when the residence was unoccupied. The ventilation air increased the datum temperature and reduced the daily compressor operation time for mean daily temperatures below 79° F. because of the "flywheel" effect of the cool night air. Above 79° F. the compressor operation time was increased because the increased load due to introduction of warm outside air during the day more than offset the "flywheel" effect of night-air cooling. A more complete analysis of night-air cooling is presented in a later section.

The lower part of Figure 14 shows the compressor operation during the 1957 cooling season. In general, the points for Series S57-3 are below the average curve. This was because Series S57-3 was conducted later in the season when there were fewer sunlight hours. The points for all series were scattered due to variations in the internal load and variations in solar radiation.

The datum temperature was 56° F. This was the lowest datum temperature of any test series conducted in Research Residence No. 2. The oven was used daily at the time of the evening meal, which was not usual during the previous investigations.

The average capacity of the cooling unit was 21,660 B.t.u.h. during S57-1 and varied from 20,900 B.t.u.h. to 23,200 B.t.u.h. during S57-2 (800 c.f.m.). The variation in average capacity was primarily due to the variations in condenser cooling water-flow rate. The water-flow rate varied from 131 g.p.h. during S57-1 to from 136 to 188 g.p.h. during S57-2. The higher water-flow rate reduced the condensing temperature and increased the unit gross capacity. The blower operated continuously during all of the 1957 series and power input to the blower was 0.49 kw., 0.43 kw., and 0.33 kw. during Series S57-1 (800 c.f.m.), S57-2 (800 c.f.m.), and S57-3 (600 c.f.m.), respectively. The net cooling capacities were reduced by the heat equivalents of the power inputs to the blower. The net capacity for Series S57-1 was 21,600 B.t.u.h. minus 1,680 B.t.u.h. or 19,980 B.t.u.h.; the net capacities for Series S57-2 were 20,900 B.t.u.h. minus 1,470 B.t.u.h. or 19,430 B.t.u.h., and 23,200 B.t.u.h. minus 1,470 B.t.u.h. or 21,730 B.t.u.h.; and the net capacity for Series S57-3 was 20,900 B.t.u.h. minus 1,330 B.t.u.h. or 19,570 B.t.u.h. Under design weather conditions of 85° F. M.D.T. (95° F. maximum dry-bulb, 75° F. minimum dry-bulb) the compressor operated 15 hours per day.

The amount of moisture removed from the residence was affected by compressor operation time and also by the method of blower operation as shown in Figure 15. For a given mean daily temperature the compressor operation time was greater with continuous than with cyclic blower operation. However, more moisture was removed with cyclic blower operation due to the fact that continuous blower operation allowed re-evaporation of the con-

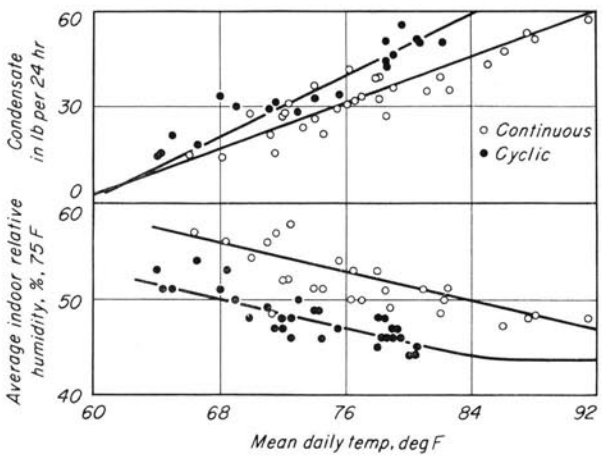


Figure 15. Effect of blower control method on daily average indoor relative humidity and moisture removed from residence

densate from the coil when the compressor was off. Cyclic blower operation allowed the moisture to remain on the coil surface while the compressor was off. With an 80° F. M.D.T., the cyclic blower operation increased the moisture removal by approximately 12 pounds per day. The curve for cyclic blower operation tends to meet the continuous blower curve at an extremely high mean daily temperature since the compressor operation would then become almost continuous.

The lower part of Figure 15 shows the effect of method of blower operation on the average relative humidity maintained in the residence. Cyclic blower operation resulted in lower relative humidities because of the removal of a greater amount of moisture per day from the house. The extrapolated segment of the cyclic blower relative humidity curve is shown as approaching the continuous blower operation curve at the higher mean daily temperature where total hours of blower operation would be similar with either method of blower operation.

The over-all static pressure loss of the small-pipe duct system external to the air conditioning unit was limited to a design value of 0.50 inch of water. The average measured static pressure at the bonnet and in the return at the furnace entrance was 0.316 and −0.062 inch of water, respectively, for an over-all external static pressure of 0.378 inch of water. The over-all loss was less than design because the system had an air-carrying capacity greater than the 560 c.f.m. total air-flow rate. In rooms such as the bedrooms where one 4-inch-diameter duct did not have sufficient capacity to carry the load of the room, the only alternative was to use two 4-inch-diameter pipes. This resulted in a lower resistance to air flow than with a single pipe sized exactly for the required air-flow rate and design pressure loss.

The ceiling diffuser supply duct system was designed for an air-flow rate of 800 c.f.m. with a bonnet static pressure of 0.15 inch of water. The single return in the entrance hall had been designed and used with the small-pipe perimeter system previously installed (560 c.f.m.). According to information contained in *Manual 4*[13] the return was adequate for 800 c.f.m. with a pressure loss of 0.05 inch of water. The average measured return static pressure was 0.105 inch of water, or more than twice the design pressure loss.

Of the design bonnet static pressure of 0.15 inch of water, 0.10 inch was utilized for the branch duct design, and the remaining 0.05 inch was utilized for the design of the riser from the cooling unit bonnet to the attic bonnet. The measured static pressures in the cooling unit bonnet and in the attic bonnet were 0.319 and 0.154 inch of water, respectively, which were much higher than the design values. The combined static pressure loss was 0.424 inch of water. Most of the error in estimating the loss from the cooling unit bonnet to the attic bonnet was due to inability to predict the loss of the fitting used to connect the 9-inch by 18-inch riser to the furnace bonnet. This fitting consisted of two short-radius elbows and was necessary because the furnace could not be moved to a position directly below the riser. In addition, a considerable pressure loss occurred between the cooling unit bonnet and the furnace bonnet, to which the riser was connected. The principle source of error in estimating the required attic bonnet static pressure was in the estimation of the pressure loss of the butt take-offs used to connect the branch ducts to the attic bonnet. In the living room east and kitchen branch ducts the measured pressure losses of the butt take-offs were 0.06 and 0.07 inch of water, respectively. In each case the take-off losses were more than half the design pressure loss allowed for the entire branch duct.

The floor register supply duct system was designed for an air-flow rate of 800 c.f.m. with a bonnet static pressure of 0.15 inch of water. The single return in the entrance hall was designed for the small-pipe perimeter system previously installed (560 c.f.m.). According to information contained in *Manual 4*[13] the return duct was adequate for 800 c.f.m. with a design static pressure loss of 0.05 inch of water, the usual value assigned to single centrally located returns. The return static pressure loss during S57-2 was 0.101 inch of water, more than twice the design pressure loss. Reducing the air-flow rate to 600 c.f.m. during S57-3 reduced the return static pressure loss to 0.057 inch of water.

The bonnet static pressure during Series S57-2 (800 c.f.m.) was 0.186 inch of water, 24% higher than the design value of 0.15. The difference was partly due to higher flow rates to the kitchen and dinette than indicated by the heat gain calculations to give good room-to-room temperature balance. To achieve good balance the branch ducts to all other rooms were dampered, resulting in higher pressure losses. When the air-flow rate was 600 c.f.m. during S57-3 the bonnet static pressure was 0.108 inch of water.

V. COMPARATIVE SUMMARY OF SYSTEMS

A. COMPARISON OF SINGLE SEASON SYSTEMS

1. Cooling

A comparison of comfort conditions during cooling should include both air motion and temperature variations. Table 8 contains a summary of the living zone (4-inch to 60-inch level) temperature variations at design conditions of 20° F. indoor-outdoor temperature difference for the three systems herein reported and also for a high sidewall system investigated[14] during the 1952 cooling season in Research Residence No. 2. Where more than one test condition was investigated with an outlet type, the test condition which resulted in more comfortable conditions is included in the table. For example, with floor registers the higher air-flow rate resulted in more comfortable conditions (Series S57-2) and, therefore, the variations obtained during this test series are listed in Table 8. The supply (face) velocity of the perimeter floor diffusers and floor registers are listed in parentheses alongside the variations for each room, because the variations were highly dependent upon the supply velocities. Supply velocities are not listed for ceiling diffusers and high sidewall registers because the variations were less dependent upon supply velocity.

Variations in the bath and north bedroom are not listed for the high sidewall system due to the fact that the supply air jet was directed onto the breathing level thermocouple producing a negative variation which was not representative.

Ceiling diffusers produced the smallest variations, a maximum of 0.6° F. in the living room and an average variation of only 0.2° F. High sidewall registers produced the next smallest variations with a maximum of 1.9° F. in the living room and an average of 1.1° F. although dropping of the supply jets caused localized drafts. Floor registers produced variations of approximately 1° F. in rooms which had high supply velocities, but the variation increased to 3.9° F. in the bath where the supply velocity was only 90 f.p.m. The variations produced by perimeter floor diffusers were the most dependent on supply velocity of all outlet types, and higher supply velocities were required to produce acceptable variations. The living zone variations were from 2.0° F. for a supply velocity of 1,100 f.p.m. to 7.2° F. for a supply velocity of 240 f.p.m.

It has been suggested in the "Proposed Standard for Comfort Air Conditioning"[15] that the variations should not exceed 1° F. per foot. This means that the living zone variations should not exceed 5° F. With this criteria all systems produced acceptable variations except the perimeter floor diffusers in the bath and north bedroom.

The supply air from ceiling diffusers was well diffused in the upper levels of the rooms and, consequently, the room air temperatures were uniform with complete absence of cooler or warmer locations. The air motion was most acceptable and no drafts occurred. When the supply velocity of the floor registers and perimeter floor diffusers was sufficient to project the cool supply air into the upper levels of the room, the room air motion and temperature variations at each level were acceptable. When the supply velocity was insufficient the cool supply air created cool areas and drafty conditions, especially near the floor. With high sidewall registers the cool supply air jets entered the living zone and created cool areas and drafty conditions.

Table 8

Comparison of Living Zone Variations During Cooling at Design Weather Conditions

Outlet Type and Location	Room Temperature Variation (4-inch to 60-inch level), °F.						
	Living Room	South Bedroom	Bath	North Bedroom	Dinette	Kitchen	Average
Perimeter Floor Diffusers	2.3(1040)*	4.7(520)	7.2(240)	5.6(530)	2.4(870)	2.0(1100)	4.0(710)
Ceiling Diffusers	0.3	0.6	−0.2	0.2	−0.1	0.2	0.2
Floor Registers	1.1(537)	1.3(429)	3.9(90)	1.7(367)	0.6(648)	1.0(706)	1.0(463)
High Sidewall Registers	1.9	1.7	**	**	0.1	0.7	1.1

* The supply velocity of the perimeter diffusers and floor registers are listed in parentheses (f.p.m.).
** The supply air jet was directed on to the breathing level thermocouple, giving a negative variation which was not representative for these rooms.

Table 9

Comparison of Systems During Heating

	Indoor-Outdoor Temperature Difference, °F.	Perimeter Floor Diffusers, Series G-4	High Sidewall, Series A-11	Floor Registers, Series L-3	Ceiling Diffusers, Series K-3
1. Average Room Air Temperature Variations, °F.					
Floor-to-Sitting Level	40	1.3	1.9	2.5	2.3
Floor-to-Breathing Level		1.8	4.2	4.6	5.5
Floor-to-Ceiling Level		2.0	9.4	5.9	...
Floor-to-Sitting Level	70	1.6	2.6	2.8	3.0
Floor-to-Breathing Level		3.5	6.9	6.5	8.2
Floor-to-Ceiling Level		3.9	17.6	8.8	...
2. Average Floor Surface Temperature, °F.	40	70.9	70.0	68.8	68.6
	70	70.4	69.3	65.3	64.8
3. Average Underside Floor Surface Temperature, °F.	40	71.7	69.2	67.9	67.2
	70	76.3	71.8	69.1	61.7
4. Average Air Temperature 4 Inches Below Floor, °F.	40	73.9	72.1	70.5	68.0
	70	76.3	71.8	69.1	61.7
5. Average Ceiling Surface Temperature, °F.	40	70.4	72.7	71.2	79.0
	70	67.5	74.5	69.6	82.7

For example, in the living room, the jets entered the living zone only a short distance from the inside wall, the location of the register, and were only 3 feet above the floor at a distance of 3 feet from the opposite wall. The air velocity was 100 f.p.m. at the latter position. The drafts caused by the jets entering the living zone were noticed by the occupants and by persons visiting the residence.

2. Heating

Table 9 contains a summary of the room air temperature variations, floor and underside floor surface temperatures, basement air temperature 4 inches below the first floor, ceiling surface temperatures, and furnace fuel consumption for the three systems herein reported and also for a high sidewall system investigated[2] during the 1948–49 heating season. Average data are listed for both average (40° F. indoor-outdoor temperature difference) and severe (70° F. indoor-outdoor temperature difference) winter weather. The series chosen for each system represented the best overall performance but it should be noted that only small differences in the items listed resulted from variations in the operating conditions.

Floor diffusers produced much smaller variations than any other system. For example, the floor-to-breathing level variation at 70° F. indoor-outdoor temperature difference was 3.5° F. for perimeter floor diffusers. The next smaller living zone differential was obtained with the floor registers, 6.5° F., followed by the high sidewall registers, 6.9° F., and the ceiling diffusers, 8.2° F. The variations produced by ceiling diffusers, floor registers, and high sidewall registers exceeded the maximum variation of 1° F. per foot of elevation suggested in the proposed standard mentioned previously. The perimeter floor diffusers also produced the warmest floor surface, warmest underside floor surfaces, and highest basement air temperature, followed by the high sidewall registers, floor registers, and ceiling diffusers. However, as was shown in the discussion of room air temperature variations, the addition of heat below the first-story floor increased floor surface temperatures and also decreased the temperature variation in the occupied zone. The warmest ceiling surface temperatures were produced by the ceiling diffusers and the coolest ceiling surface temperatures were produced by the perimeter floor diffusers.

The floor surface temperatures, underside floor surface temperatures, and 4-inch below-floor air temperatures decreased from a maximum with the floor diffuser system to a minimum with the ceiling diffuser system and the items are listed in that order, by systems, in the table. The greater heat loss from the furnace and duct work located in the basement was not completely lost but increased the comfort conditions within the conditioned space. Also, the floor to sitting level differentials increased in the same order, suggesting that the temperature differentials in the lower levels of the occupied space were largely dependent upon the floor surface temperatures and more or less independent of the method of heat addition, a characteristic that has been noticed in previous investigations.[2, 3] The differentials above the sitting level were primarily dependent on the method of heat addition with minor differences resulting from differences in operating procedures, i.e., air-flow rate, temperature rise, and firing rate, as noted in the respective sections.

B. SELECTION OF A YEAR-AROUND SYSTEM

Choice of the correct type of air distribution system for year-around air conditioning for a specific installation depends on an evaluation of many factors: whether the principal need is for heating or for cooling, design weather conditions, frequency of occurrence of design conditions, average weather conditions, the thermal protection of the home in which the system is to be installed, and preference of the homeowner regarding location of outlets.

The floor diffusers, floor registers, and ceiling diffusers were capable of introducing large amounts of air into the conditioned space without producing excessive drafts. This was also true of the high sidewall outlets during heating, but during cooling the supply jets dropped into the occupied zone. The single, centrally located return air grille performed satisfactorily with all systems, and the use of this type of return may reduce the cost of installation considerably. In larger homes it may be desirable to install more than one return air grille, each to serve a different area. The thermal protection of Research Residence No. 2 was good, resulting in a heat loss of only 30 B.t.u.h. per square foot of floor area, but the walls, the primary source of drafts, had a large percentage of glass area which tended to reduce the effectiveness of the insulated wall. For example, in the living room the window and door accounted for 54% of the total south exposed wall area. The average heat transmission factor for this wall was 0.29 B.t.u.h. per square foot. If the wall had consisted of 25% window and door area and had only 2-inch thick insulation in the wall, the average heat transmission factor would have been reduced to 0.19 B.t.u.h. per square foot. Therefore, the results reported in this bulletin are reasonable for most present-day residences.

If heating, rather than cooling, is the primary need the perimeter system should be installed. The performance of this system was good regardless of the air-flow rate utilized and is therefore applicable for cooling which may be the year-around air-flow rate. The primary consideration in the design of the duct system is to have an adequate supply velocity during cooling, indicating that the diffuser sizes should be based on the total cooling air-flow rate apportioned in accordance with the heat-gain calculations. The supply velocity should be a minimum of 500 f.p.m., and 600 or 700 f.p.m. should result in better performance. Floor diffusers are but one of a group of perimeter-outlet types, all of which produce similar air patterns. Basically, similar results should occur whether the perimeter outlet is a floor diffuser, low sidewall diffuser, baseboard diffuser, or extended baseboard diffuser, as long as the discharge pattern is similar to the spreading pattern of the floor diffusers.

If cooling is the primary need and heating is only a secondary need in an area with a mild winter, design temperature and the average winter temperature is above 40° F., the ceiling diffuser system should be given first consideration because of its much better performance during cooling.

If either a high sidewall or a floor register system is already installed in a residence for heating, and cooling is to be added, either of these systems will perform acceptably. With a high sidewall system the critical characteristic is the region where the cool supply air enters the living zone. With a horizontal discharge, cool areas will result because of this characteristic. A possible preventative measure is to install high sidewall registers with adjustable horizontal vanes and to direct the supply air upward and spread it over the ceiling to more completely diffuse the cool air before it enters the living zone. With a floor register system it is necessary to provide a minimum supply velocity during cooling of 500 f.p.m. If the supply velocity is less than 500 f.p.m. there is a good possibility that the cool supply air will fail to reach the upper levels of the rooms. Instead, the cool air may fall to floor level and create cool, drafty areas.

It should be mentioned that any heating system will produce more comfortable living conditions if the floors are warm. Therefore, addition of heat to the underside of floors during the heating season with the systems which perform poorly during heating will improve the comfort conditions and this may at times be the best compromise. In some instances it may be advisable to insulate the underside of the floors to reduce the heat loss through them.[16]

VI. TEMPERATURE BALANCE

A. GENERAL

One of the problems associated with utilizing a single duct system for year-around air conditioning, both heating and cooling, is that each room usually requires a different proportion of the total air-flow rate for cooling than it does for heating. Solar gain during cooling is the primary reason for the difference. Consequently, some adjustment of dampers must be made at the time of change over from cooling to heating or vice versa to avoid excessive temperature differences between rooms. Special studies were conducted to determine the unbalance that would result when a system that had been adjusted for cooling balance was utilized for heating and vice versa. The investigation was extended to determine simple and effective means of reducing the unbalance.

B. CALCULATED AIR CONDITIONING LOADS

The calculated heat gains and heat losses for design conditions are listed in Table 1. The ratio of the calculated load of each room to the total calculated load of the residence is shown in Table 10 for both cooling and heating. The last column in Table 10 lists the cooling/heating load ratios for each room obtained by dividing the cooling load ratio by the heating load ratio. The cooling/heating load ratios indicate the theoretical change in the air-flow rate required for cooling in relation to the air-flow rate required for heating. For example, the cooling/heating load ratio of the living room was 0.95, indicating that the air-flow rate to the living room should have been decreased by 5% when changing from heating to cooling. The cooling/heating load ratios for the entire house ranged from 0.89 to 1.21.

C. ACTUAL AIR-FLOW RATES FOR TEMPERATURE BALANCE

Each of the three systems investigated was balanced by means of the branch duct dampers during one or more test series. The balance was considered satisfactory if the maximum room-to-room temperature difference was less than 2° F. on an average day. The ratios of the room air-flow rates to the total air-flow rates for both cooling and heating are listed in Table 11. The last column in the table lists the cooling/heating air-flow ratios obtained by dividing the cooling air-flow ratios by the heating air-flow ratios. The cooling/heating air-flow ratios indicate the actual increase in the room air-flow rate required for cooling in relation to the air-flow rate required for heating. For example, the cooling/heating air-flow ratio for the living room was 1.22, indicating that the air-flow rate to the living room had to be increased 22% when changing from heating to cooling in order to maintain a satisfactory room-to-room temperature balance. The cooling/heating air-flow ratios varied from 0.83 to 1.44.

The calculated cooling/heating load ratios are graphically compared with the cooling/heating air-flow ratios in Figure 16. The upper bars represent the calculated load ratios and the lower bars represent the air-flow ratios. A value of unity would represent no difference in the cooling and heating load or air-flow ratios. The calculated load ratios did not provide an accurate indication of the air-flow ratios required for room-to-room temperature balance. The three rooms, the two bedrooms and bath, whose calculated load ratios indicated that more air would be required for cooling than for heating actually required less and the converse was

Table 10

Ratio of Calculated Room Load to Total Load

Room	Cooling	Heating	Cooling/Heating
Living Room	.335	.352	0.95
South Bedroom	.200	.180	1.11
Bath	.057	.047	1.21
North Bedroom	.196	.181	1.08
Kitchen-Dinnette	.213	.240	0.89

Table 11

Ratio of Room Air-Flow Rate to Total Air-Flow Rate

Room	Cooling	Heating	Cooling/Heating
Living Room	.365	.300	1.22
South Bedroom	.166	.200	0.83
Bath	.035	.085	0.41
North Bedroom	.157	.215	0.73
Kitchen-Dinette	.279	.194	1.44

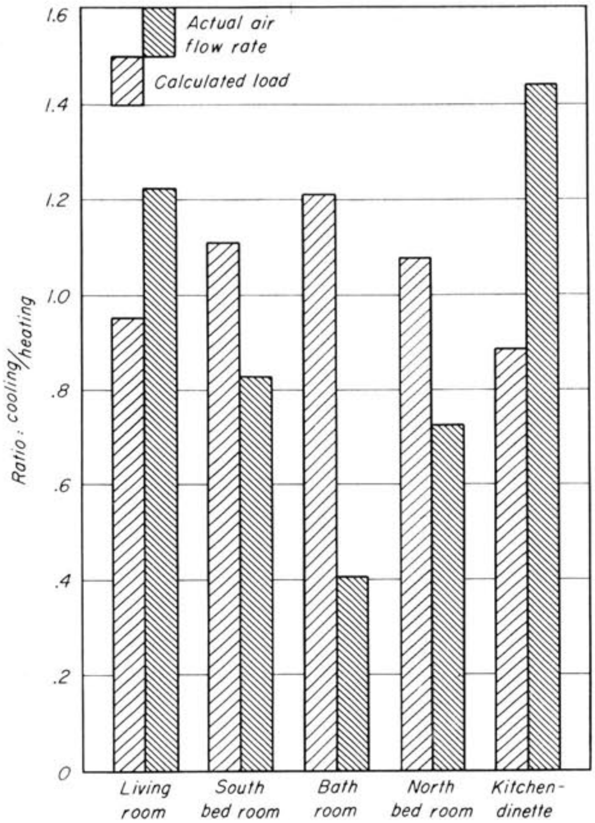


Figure 16. Comparison of calculated load ratios and actual air-flow ratios for year-around air conditioning

true for the living room and kitchen-dinette. Because the doors between rooms remained open and because of the location of the return grille, the residence was, in effect, divided into two zones, the east zone consisting of the living room and kitchen-dinette and the west zone consisting of the bedrooms and bath. The calculated load ratios indicated that the air-flow rate to the east zone should be decreased 7% when changing from heating to cooling. Instead, the air-flow rate to the east zone was increased 30%. Three items, in addition to inaccuracies in the load calculations, are primarily responsible for the discrepancies: heat gain or loss from ducts, heat transmission through the floor, and effects of occupancy.

The maximum difference in the cooling effect of the conditioned air supplied to the different rooms due to temperature rise in ducts was approximately 25%. The maximum difference in the heating effect of the conditioned air due to temperature drop in ducts was approximately 40%. Therefore, temperature drop in ducts had a larger effect on balance than did temperature rise, and the rooms farther from the bonnet — bedrooms and bath — required a larger percentage of the total air flow for heating than for cooling.

During cooling the floor was a neutral surface of uniform temperature. During heating the floor surface temperatures varied widely as was shown by the floor surface isotherms. Considerable heat regain occurred through the floors in the area above the furnace, primarily in the dinette and living room, and little or no heat regain occurred through the floor of the bedrooms. The effect was similar to the effect of duct heat transfer, the air requirements for heating the west zone were increased.

The largest load due to occupancy was from appliance-released heat which increased the cooling load of the kitchen. Although the temperature readings upon which the cooling balance was based were obtained at 3:00 p.m. there was probably a "flywheel" effect from earlier use of the kitchen. Because the temperature readings upon which the heating balance was based were obtained at 6:30 a.m., the effects of occupancy were negligible.

D. BALANCING METHODS

It was learned that the perimeter system could be satisfactorily balanced during heating with the kitchen diffuser completely closed. During cooling it was necessary to utilize the kitchen diffuser. Some attempts to obtain year-around balance were based on this characteristic. When a system adjusted for cooling balance with the kitchen diffuser open was utilized for heating, the maximum temperature difference between rooms was approximately 4.5° F. Closing the kitchen diffuser decreased the maximum temperature difference between rooms to approximately 3° F. On the other hand, when a system adjusted for heating balance with the kitchen diffuser closed was utilized for cooling, the maximum temperature difference between rooms was approximately 4° F., and opening the kitchen diffuser decreased the unbalance by only 0.5° F. Although the air-flow rate to the kitchen-dinette was increased by 10% of the total air-flow rate when the kitchen diffuser was opened, the living room flow decreased, resulting in a net increase to the east zone of only 6% of the total air-flow rate. This amount was insufficient to correct the unbalance. The investigation revealed that while some improvement in balance was achieved by opening or closing a single outlet the effect was not sufficient to correct the unbalance because of the interdependence of outlets in extended plenum type duct systems. A more effective method of obtaining year-around balance would be to balance the basic system for either heating or cooling, depending upon which was more important in the specific locality, and then to add additional ducts from the bonnet to be utilized in areas of unbalance.

Air-flow rate and the accompanying air temperature rise during heating had a negligible effect on the room-to-room temperature balance.

During the 1955–56 heating investigation an auxiliary electric resistance heater was installed in the living room to simulate a reduced living room heat loss. Heat was added at a rate of 3,640 B.t.u.h., which was equal to 38% of the calculated design heat loss of the living room. This produced a simulated condition wherein the calculated heat loss of the living room was 5,870 B.t.u.h. instead of its true value of 9,510 B.t.u.h. Therefore, the living room cooling/heating load ratio was increased. It had already been determined that the system could be balanced during heating with the kitchen diffuser closed. The reduced heat loss of the living room permitted the living room east diffuser to also be closed during heating. The duct system was the same as the floor diffuser system shown in Figure 3 except that the dinette branch duct take-off was transferred from the east extended plenum to the cooling unit bonnet. This permitted installation of a damper at the entrance of the east extended plenum so that the kitchen and living room east diffusers could be utilized for cooling and easily closed for heating. During Test Series G-11 the system was adjusted for good balance during cooling (same as during S54-1) with the living room east and kitchen diffusers utilized. The maximum room-to-room temperature difference was 5.7° F. During Series G-12 the damper at the entrance of the east extended plenum was closed and the maximum temperature difference between rooms was reduced to 2.3° F. Therefore, where a large cooling/heating load ratio exists an effective method of correcting the unbalance is to group the ducts required for only one season so that they may be controlled by a single damper.

VII. TEMPERATURE DROP AND TEMPERATURE RISE IN DUCTS

A. TEMPERATURE DROP

A procedure for the calculation of temperature drop in ducts was developed in University of Illinois Engineering Experiment Station Bulletin No. 351.[17] Heat transfer coefficients as determined in the laboratory compared favorably with the values for the coefficients published in the literature. It was concluded that temperature drop in ducts could be calculated and that an elaborate laboratory set up with controlled ambient temperatures to measure temperature drop in ducts was not necessary. Temperature drop in ducts of 6-, 8-, 10-, 12-, 16-, 20-, 24- and 30-inch diameter were calculated for air velocities of 400, 600, 800, and 1,000 f.p.m.

In an actual installation the question arises as to how to estimate temperature drop in a system of ducts which includes an extended plenum or trunk duct and branch ducts. The residence with duct work in the attic and in the basement equipped with thermocouple grids provided an installation in which calculated temperature drop could be compared with the measured temperature drop.

The burner operated continuously on several occasions during Series G-3, 1954–55 heating season (floor diffusers), when outdoor conditions were near design. On one occasion the burner operated continuously from 7:40 p.m. until 10:10 a.m. A comparison of the measured and calculated temperature drop from bonnet to diffuser is contained in Table 12. The temperature measurements were made at 12:24 a.m. at which time equilibrium conditions had been attained. The measured temperature drop, the difference between the bonnet temperature and the temperature at each diffuser, includes both the temperature drop in the branch and trunk duct. The calculated temperature drop is based on information given in Bulletin 351 and was obtained by extrapolation, since 4-inch diameter ducts were not considered in the bulletin. The calculated temperature drop was determined by assuming a constant-diameter duct from bonnet to the diffuser rather than the extended plenum and branch ducts installed. Calculated temperature drops were obtained for five of nine diffusers. The air-flow rates for the other four branches were below the velocity range covered in Bulletin 351.

Excellent agreement was obtained between the calculated and measured temperature drops for the living room (W), south bedroom (W), and north bedroom (W) ducts, all of which were connected to the west extended plenum. Poor agreement was obtained for the living room (E) and dinette ducts, which were connected to the east extended plenum. The large deviation for the ducts connected to the east plenum was primarily due to the heat loss which occurred as the heated air passed through the large bonnet above the cooling unit enroute from the heating bonnet to the entrance of the east extended plenum. When the air temperature at the entrance to the east extended plenum was used to calculate the temperature drop good agreement was

Table 12

Comparison of Measured and Calculated Temperature Drop in Ducts During Continuous Burner Operation

Series G-2

Diffuser Location	Measured Length, ft.[a]	Air Flow Rate, Standard c.f.m.	Velocity, f.p.m.	Temperature Drop, °F. Measured	Temperature Drop, °F. Calculated	Deviation from Measurement, %	Error, % of Capacity
Living Room (E)	21	52	595	36.6	24.1	−34.2	14.2
Living Room (W)	19	47	538	24.5	25.2	+2.9	0.8
South Bedroom (S)	30	22	252	40.9	...	...	...
South Bedroom (W)	30	36	412	40.5	41.4	+2.2	1.8
Bath	14	12	137	32.9	...	...	...
North Bedroom (W)	23	42	481	32.9	31.3	−0.6	1.8
North Bedroom (N)	25	23	264	43.0	...	...	...
Dinette	10	52	595	27.0	12.4	−54.0	16.5
Kitchen	16	5	157	57.4	...	...	...

[a] Length from furnace plenum to diffuser.
Note: The bonnet temperature was 158.8° F. and the return air temperature was 70.6° F.

Table 13

Temperature Drop in Attic Ducts

Series K-1
Outdoor Air Temperature = −2.8° F.
Attic Air Temperature = 15.2° F.

Return Air Temperature = 68.4° F.
Avg. Supply Air Temperature = 104.5° F.
Burner on 25 min.
Total Air-Flow Rate = 789 c.f.m.

Run	Duct Diameter, in.	Length of Run, ft.	Vapor Barrier	Velocity, f.p.m.	Entering Temperature, °F.	Leaving Temperature, °F.	Temperature Drop	
							°F.	°F./ft.
NBR	7	17	Paper	653	103.2	101.0	2.2	0.13
SBR	7	15	Aluminum Foil	568	103.8	101.2	2.6	0.17
LRE	7	10	Aluminum Foil	402	101.9	99.1	2.8	0.28
LRW	7	8	Paper	399	104.9	103.0	1.9	0.24
Din	6	4½	Paper	450	101.9	101.2	0.7	0.16
Kit	6	11	Aluminum Foil	346	100.8	97.6	3.2	0.29
Bath	5	13	Paper	684	104.5	101.2	3.3	0.25

obtained. The maximum deviation of the calculated temperature drop from the measured temperature drop, 54.0%, represented an error of 16.5% of the total heating capacity. The deviation of the calculated temperature drop from the measured temperature drop for the three ducts connected to the west extended plenum represented 0.8, 1.0, and 1.8% of the total heating capacity.

This analysis indicates that a good approximation of duct heat loss results from assuming that the heat loss from ducts in an extended plenum system is the same as for an individual duct system; although the accuracy undoubtedly depends upon the ratio of branch velocity to plenum velocity and the ratio of branch length to total length.

Table 13 contains a summary of the duct air temperatures measured on a day during Series K-1 (ceiling diffuser system) when the outdoor air temperature was −2.8° F., the total air-flow rate was 789 c.f.m., the attic air temperature was 15.2° F., and the burner had operated continuously for 25 minutes. The duct air temperatures listed were measured in the attic branches. The data are grouped according to duct diameter and listed within each size grouping in order of decreasing duct air velocities. In general, higher duct air velocities resulted in lower temperature drop per foot of duct. The maximum temperature drop was 0.29° F. per foot and the minimum temperature drop was 0.13° F. per foot. For this example the heat loss from the attic branch ducts was 6.6% of bonnet capacity. Heat loss from the duct work between the furnace bonnet and the attic bonnet take-offs represented 3.6% of bonnet capacity for a total loss of 10.2% of bonnet capacity.

At lower air-flow rates the duct air temperatures were higher and the duct air velocities were lower and, therefore, the heat losses from the ducts were larger although data are not available for a complete comparison. On a day during Series K-3 when the outdoor air temperature was 17.1° F. and the burner had operated for approximately 10 minutes the over-all duct heat loss was 16.9% of bonnet capacity. The loss from the branch ducts located in the attic was 11.2%, and the remaining 5.7% was lost from duct work between the furnace bonnet and the attic bonnet take-offs.

The heat lost from the attic ducts was not available for warming floors as was the case with the basement duct work. Since the losses were as much as 11% of bonnet capacity even with 2 inches of insulation on the duct work, the importance of proper insulation is indicated. Without duct insulation it is doubtful that the house could have been adequately heated.

No extended period of continuous burner operation occurred during the investigation of the floor register system. However, it was possible to obtain sufficient data that the heat loss from the duct work could be approximately determined. On a day during Series L-1 (800 c.f.m.) when the outdoor air temperature was 4.2° F. and the burner had operated continuously for 10 minutes, the heat loss from ducts was 15.5% of bonnet capacity. During L-2 (600 c.f.m.) when the outdoor air temperature was −5.8° F. and the burner had operated continuously for 40 minutes the heat loss from ducts was 17.9% of bonnet capacity. Ten minutes was not sufficient time for equilibrium conditions to be established and the heat loss was larger than it would have been after equilibrium temperatures had been established. A larger difference in the heat loss from ducts than given above would have been measured if both systems had been under equilibrium conditions.

Heat losses from the floor register system were also available for warming the first-story floor and the basement.

Table 14

Comparison of Measured and Calculated Temperature Rise in Basement Ducts During Continuous Compressor Operation

Series S54-1

Diffuser	Measured Length, ft.[a]	Air Flow Rate, Standard c.f.m.	Velocity, f.p.m.	Measured Diffuser Temperature, °F.	Temperature Rise, °F.		Deviation of Calculated from Measured, %	Error, % of Sensible Capacity
					Measured	Calculated		
Living Room (E)	21	92	1060	58.9	4.2	3.85	−8.33	1.64
Living Room (W)	19	91	1040	59.6	4.9	3.57	−27.14	6.24
South Bedroom (S)	30	36	410	62.3	7.6	9.00	18.40	6.57
South Bedroom (W)	30	54	620	61.6	6.9	7.54	9.28	3.00
Bath	14	21	240	60.1	5.4	...	...	...
North Bedroom (W)	23	63	720	59.6	4.9	5.47	11.63	2.68
North Bedroom (N)	25	30	340	62.0	7.3	...	...	...
Dinette	10	76	870	56.8	2.1	2.24	6.67	0.66
Kitchen	16	96	1100	58.5	3.8	2.90	−23.68	4.23

[a] Length from furnace plenum to diffuser. Temperature rise calculated as if branch ducts were connected to air conditioner plenum rather than the extended plenum with branch ducts installed.

Note: All branch ducts 4-inch diameter.
Basement air temperature 78.5° F.
Bonnet temperature 54.7° F.
Return air temperature 76.0° F.

B. TEMPERATURE RISE

A comparison of the measured and calculated temperature rise of conditioned air from the bonnet of the summer air conditioner to the diffuser in the floor diffuser system is given in Table 14. The data were obtained at 3:00 p.m. during a day in Series S54-1, 1954 cooling season. The compressor operated continuously from 9:30 a.m. until 11:15 p.m. The measured temperature drop was obtained by subtracting the bonnet temperature from the diffuser temperature and includes the temperature rise in both the branch and trunk ducts. The calculated temperature rise is based upon information contained in the Appendix, "Calculated Temperature Rise in Circular Ducts." The duct system was assumed to consist of constant diameter ducts extending from the bonnet to the diffuser rather than the extended plenums and branch ducts which were installed. Calculated temperature rise values were obtained for seven of nine branch ducts. The velocities in the bath and north bedroom (N) branches were below the range covered in the Appendix. The calculations were obtained from the entering temperature–temperature rise curves, and corrections were made for velocity and ambient temperature, as explained in the Appendix. The basement air temperature was 78.5° F.

The deviation between the measured and calculated temperature rise varied from +18.40 to −27.14%, based on the measured temperature rise. The measured temperatures were accurate within ±0.3° F. This error could result in an error of 0.6° F. in temperature rise which could reduce the maximum deviation to only 14.9%. There were other possible sources of error: the assumption of an individual duct system, the assumption that the basement temperature was uniform, the assumption that the mean radiant temperature and the ambient air temperatures were equal, the assumption in the calculations that condensation did not occur whereas a small amount of condensation did occur on the bonnet and for a short distance on the extended plenums.

Although the variations listed above occurred, the maximum error based on the total drop of temperature through the cooling unit, which is a measure of the sensible capacity, was only 6.57%. The average error was 3.4% of the sensible capacity. The heat gain to ducts represented 22.0% of the total sensible capacity of the cooling unit.

The attic duct work used with the ceiling diffuser system was insulated with 2-inch-thick glass fiber insulation which was covered with a vapor barrier. The vapor barrier for three branch ducts was aluminum foil and for the other four branch ducts was asphalt-impregnated paper. The maximum attic air temperature occurred about 2:00 p.m., 2 hours after the maximum roof surface temperature and 1 hour before the maximum heat gain to the conditioned space. Because of the thermal lag of the duct insulation the maximum heat gain to the ducts occurred after the maximum attic air temperature, although the lag was not determined. Table 15 contains a comparison of the measured and calculated temperature rise in the attic ducts at 3:00 p.m. on a day when the outdoor air temperature was 90° F., the attic air temperature was 120° F., and the average temperature of the attic surfaces was 122° F. Calculated temperature rise values were obtained for five of seven branch ducts. The velocities in the bath and dinette branches were below the range covered in the Appendix. The cal-

Table 15

Comparison of Measured and Calculated Temperature Rise in Attic Ducts During Continuous Compressor Operation (Series S57-1)

Diffuser Location	Duct Diameter, in.	Measured Length, ft.	Vapor Barrier	Air-Flow Rate, c.f.m.	Velocity f.p.m.	Entering Air Temperature, °F.	Temperature Rise, °F. Measured	Temperature Rise, °F. Calculated	Deviation of Calculated from Measured, %	Error, % of Sensible Capacity
Living Room (E)	7	10	Aluminum Foil	146	546	58.5	1.1	0.93	−15.5	1.04
Living Room (W)	7	8	Paper	179	669	58.7	0.9	0.68	−24.4	1.35
South Bedroom	7	15	Aluminum Foil	134	501	58.5	1.8	1.51	−16.1	1.78
Bath	5	13	Paper	38	278	58.5	5.5	...	...	...
North Bedroom	7	17	Paper	129	484	58.7	2.5	1.85	−26.0	3.99
Dinette	6	4½	Paper	50	257	59.2	0.4	...	...	...
Kitchen	6	11	Aluminum Foil	126	640	58.2	1.4	1.07	−23.6	2.20

Note: Attic air temperature 120° F.
Return air temperature 73.8° F.
Bonnet air temperature 57.5° F.

culations were obtained from the entering temperature–temperature rise curves and corrections were made for velocity and ambient temperature, as explained in the Appendix.

The deviation between the measured and calculated temperature rise varied from −15.5 to −26.0%, but the maximum variation in degrees Fahrenheit was only 0.65° F. The measured temperatures were accurate within ±0.3° F., which could result in most of the deviation. However, since the calculated temperature rise was always less than the measured value the deviation was probably due to other factors. As mentioned previously, the thermal lag of the insulation would increase the temperature rise to a value higher than obtained under steady-state operation at the temperature measured at 3:00 p.m. The maximum attic temperature was 123° F. at 1:45 p.m. If the calculated temperature rise had been based upon the maximum attic temperature the maximum deviation of 26% would have been reduced to 18.0%. There were other possible sources of error; the assumption that the attic temperature was uniform and the assumption that the mean radiant temperature and ambient air temperature were equal, although the small difference in average surface temperature and ambient temperature indicated that this was a reasonable assumption.

The maximum error based on the total drop of temperature through the cooling unit, which is a measure of the sensible capacity, was only 3.99%. The average error was 1.96% of the sensible capacity. The heat gain to ducts represented 17.8% reduction in the sensible capacity of which 6.8% occurred in the attic ducts and 11.0% occurred between the air cooling unit bonnet and the attic bonnet take-offs.

In the floor register system the branch ducts were short and the temperature rise through them was negligible. Practically all the temperature rise occurred between the cooling unit and the extended plenum take-offs. During continuous compressor operation the temperature rise in the east and west extended plenums was 2° F. and 3° F., respectively. The average temperature rise between the bonnet and the diffusers during continuous compressor operation was 1.6° F. for 800 c.f.m. air-flow rate and 1.8° F. for 600 c.f.m. The average temperature rise represented 9.4 and 10.2% of the sensible cooling capacity at 600 and 800 c.f.m., respectively.

VIII. COMPARISON OF AIR- AND WATER-COOLED CONDENSING UNITS

A. DESCRIPTION OF AIR-COOLED UNIT

During the 1955 cooling season investigation an air-cooled mechanical refrigeration unit was installed in the residence for comparison of its operating characteristics with the operating characteristics of the water-cooled refrigeration unit used during previous and subsequent investigations. The air-cooled unit was similar to, and made by the manufacturer of, the water-cooled unit which was described in Chapter 2, Section B. The evaporator coil was mounted in the same position. The same blower and motor were mounted above the evaporator of each unit. The evaporators of the two units were not identical. The evaporator of the air-cooled unit had a face area of 1.75 square feet and was constructed of $\frac{3}{8}$-inch copper tubing spaced 4 rows deep and 12 rows wide. The evaporator of the water-cooled unit had a face area of 1.86 square feet and was constructed of $\frac{1}{2}$-inch copper tubes spaced 3 rows deep and 9 rows wide. The fins of the evaporator of the air-cooled unit were spaced 8 per inch along the tubes. The evaporator tubes and the aluminum fins had been bonded by hydraulically expanding the tubes with the fins in position. The flow of refrigerant (Refrigerant 22) to the evaporator coil was throttled by capillary tubes. The hermetically sealed compressor was driven by a 230-volt, single-phase, 2-h.p. motor which was cooled by the suction gases. The air-cooled condenser, which was located in a compartment insulated from the evaporator, was of the finned-tube type with 6 fins per inch and had a face area of 1.94 square foot. The condenser blower had a 10-inch-diameter wheel and was driven by a $\frac{1}{4}$-h.p. motor. The blower exit was connected to the condenser by a canvas transition piece to minimize noise transmission.

The cabinet which enclosed the air-cooled condenser and the cooling air inlet and outlet ducts is shown in Figure 17. The cooling air inlet (lower) and outlet (upper) ducts were 14 inches by 20 inches, insulated with 1-inch thick blanket type vegetable fiber insulation. The cooling air intake was located in a window-well on the north side of the residence and the exhaust was above the intake, 3 feet above grade, as shown in Figure 18. The

Figure 17. Air-cooled refrigeration unit and condenser cooling air ducts

Figure 18. Cooling air exhaust weatherhead

weather heads on the inlet and exhaust were covered with 3/8-inch wire mesh.

The rated capacity of the air-cooled refrigeration unit was 22,500 B.t.u.h. at ASRE inlet air conditions with 800 c.f.m. standard air-flow rate over the evaporator and 1,400 c.f.m. standard air-flow rate over the condenser and with 95° F. air entering the condenser.

The capacity of the water-cooled unit varied from 15,600 B.t.u.h. to 19,000 B.t.u.h., depending upon the length of operation. The average capacity was 17,400 B.t.u.h. The capacity of the air-cooled unit decreased with increasing outdoor air temperature. The average capacity at 72° F. M.D.T. was 20,500 B.t.u.h. and decreased to 16,500 B.t.u.h. at 84° F. M.D.T.

B. COMPARISON OF EFFICIENCIES

The efficiency of operation with both types of condenser cooling was a function of outdoor temperatures. Figure 19a shows the effect of mean daily temperature on the heat removal per kilowatt-hour electrical input to the compressor and condenser blower motors. The heat removed per kilowatt-hour for the water-cooled unit increased from 6,400 B.t.u. per kw.h. at 68° F. M.D.T. to 7,500 B.t.u. per kw.h. at 86° F. M.D.T. as the heat removed per kilowatt-hour for the air-cooled unit decreased from 8,200 B.t.u. per kw.h. to 5,800 B.t.u. per kw.h. over the same temperature range.

C. COMPRESSOR OPERATION TIME

A comparison of the compressor operation time for the air- and water-cooled condensing units is shown in Figure 19b where the operation time is a function of the mean daily temperature. The relatively small difference in the operation time was due to small differences in the capacities of the two units, small differences in weather, and to leakage of condenser cooling air into the evaporator section. In regard to the last item, leakage of air from the condensing section into the evaporator section, an examination of the air-cooled unit revealed dust and small debris in the compressor compartment. Also, the sheet metal plates used to seal the evaporator from the compressor compartment were rusted, a condition that had not existed during previous investigations, indicating that moist outdoor air had leaked from the condenser compartment into the compressor compartment. The air could

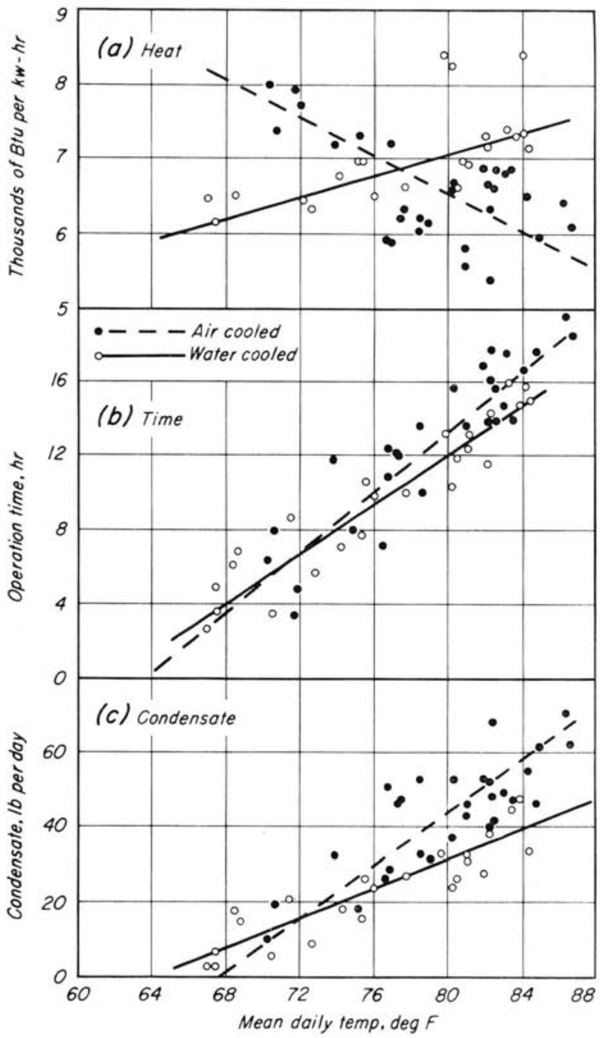


Figure 19. Comparison of water-cooled and air-cooled operation

then leak into the evaporator section. Infiltration studies[18] conducted during this investigation indicated that air leakage had occurred between the condenser and evaporator. The difference in compressor operation time was zero at 71° F. M.D.T. and 11% at 84° F. M.D.T.

D. COMPARISON OF MOISTURE REMOVED FROM RESIDENCE

The net moisture removed from the residence is shown as a function of mean daily temperature in Figure 19c for both air- and water-cooled condensing units. Approximately 50% more moisture was removed from the residence by the air-cooled unit than by the water-cooled unit. Three factors were primarily responsible for the differences. First, the

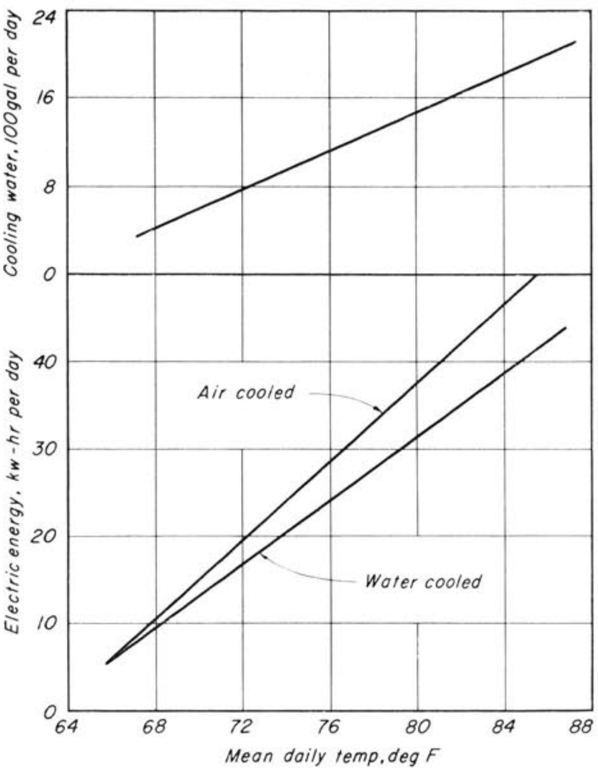


Figure 20. Condenser cooling water consumption and electrical energy consumption

evaporator coil of the air-cooled unit contained four rows of coils whereas the evaporator coil of the water-cooled unit contained only three rows. The extra row of coils decreased the bypass factor. Second, the air-cooled unit evaporator coil operated at approximately 4° F. lower temperature than the water-cooled evaporator coil. Third, the leakage of condenser cooling air introduced more humid air into the system at higher mean daily temperatures.

E. COMPARISON OF COST OF OPERATION

The electrical energy consumption for each type of refrigeration unit is shown in Figure 20 as a function of mean daily temperature and also the water consumption of the water-cooled unit is shown as a function of mean daily temperature. The electrical energy consumption includes the energy required for the condenser fan but the energy required for the recirculation blowers (8 kw.h. per day during this investigation) is not included for either unit because of its dependence upon duct design and quantity of air recirculated.

The cost of operation of the water-cooled unit may be calculated by the equation:

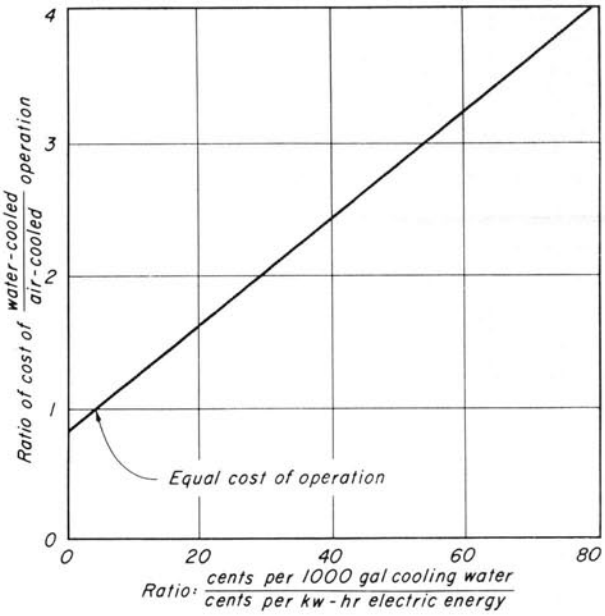


Figure 21. Comparison of operating cost of water- and air-cooling units

$$C = \frac{E\,K}{100} + \left[\frac{R}{100}\right]\left[\frac{Q}{1000}\right] \qquad (1)$$

where C = cost of operation, dollars
E = electrical energy consumed, kw.h.
K = cost of electrical energy, cents per kw.h.
Q = water consumed, gallons
R = cost of water, cents per 1,000 gallons

For air-cooled operation; $Q = O$ and $C = \frac{E\,K}{100}$ **(2)**

Dividing (1) by (2) and adding subscripts provides a convenient means of comparing the cost of operation of the two units:

$$\frac{C_{\text{water}}}{C_{\text{air}}} = \frac{E_{\text{water}}}{E_{\text{air}}} + \left[\frac{R}{K}\right]\left[\frac{Q}{1000\,E_{\text{air}}}\right] \qquad (3)$$

The ratio of $E_{\text{water}}/E_{\text{air}}$ during this investigation ranged from 1.0 at 67° F. M.D.T. to 0.80 at design conditions of 85° F. M.D.T. (maximum 95° F., minimum 75° F.). At 75° F. M.D.T. the ratio was 0.83. The value of Q/E_{air} was 40 over the entire range of weather conditions. Substituting these values into Equation 3:

$$\frac{C_{\text{water}}}{C_{\text{air}}} = 0.83 + 0.040\,\frac{R}{K} \qquad (4)$$

Equation 4 is plotted in Figure 21 where the ratio of cost of operation is shown as a function of the ratio of water cost to electrical energy costs, R/K. For the units investigated, equal cost of operation would have resulted for a ratio of R/K of 4.25. For an electrical energy rate of 2.5 cents per kw.h., approximately the national average, water costs could not have exceeded 10.6 cents per 1,000 gallons without resulting in larger operating costs for the water-cooled unit than the air-cooled unit under the assumed conditions.

Data listed in "Predicting Operating Costs for Residential Air Conditioning"[19] indicate that the values of E_{water}/E_{air} used above are representative for these types of cooling equipment.

The comparison of costs of operation is valid only for the assumed conditions. If a cooling tower is utilized to conserve water or if the cooling water inlet and outlet temperatures are different from the conditions during this investigation, the comparison of Figure 21 and Equation 4 will not be accurate.

IX. NIGHT-AIR COOLING TO REDUCE OPERATING COSTS

A. OBJECTIVE AND OPERATING PROCEDURE

During the latter part of the 1955 cooling season investigation a test series was conducted to determine the reduction in operating costs resulting from ceasing operation of the mechanical refrigeration unit and opening windows during cool nights. This procedure was called "night-air cooling." The windows and doors were opened when the outdoor dry-bulb temperature dropped to 75° F., provided that this temperature occurred before 9:00 p.m.; and were closed at 5:30 a.m. the following day. If the outdoor air was warmer than 75° F at 9:00 p.m. the windows and doors remained closed throughout the night. The air-cooled unit was utilized during this test series. Therefore, comparisons will be made with the information contained in the section on the air-cooled condensing unit.

B. EFFECT ON COMFORT CONDITIONS

The upper section of Figure 22 is a representation of the indoor and outdoor air temperature cycles. The outdoor air temperature fluctuated as the day progressed in an approximately sinusoidal curve, as shown. The indoor air temperature remained constant throughout the day until the windows and doors were opened. The windows and doors were opened when the indoor and outdoor temperatures were equal (75° F.). After the windows and doors were opened the indoor and outdoor air temperatures decreased, though not at the same rate. The outdoor air temperature decreased more rapidly than the indoor air temperature and attained a lower minimum temperature. A graphical representation of the drop of indoor air temperature after opening windows and doors versus the drop of outdoor air temperature is shown in the lower part of Figure 22. The drop of outdoor air temperature was always larger than the drop of indoor air temperature because of the heat storage of structural components of the residence. The theoretical maximum drop of indoor air temperature would be equal to the drop of outdoor air temperature, as indicated. The actual drop of indoor air temperature was 5° to 6° F. less than the drop of outdoor air temperature and a 5° F. drop of outdoor air temperature occurred before any drop of indoor air temperature. When the outdoor air temperature drop was large the indoor air dropped to an uncomfortably low temperature. For example, when the outdoor air temperature dropped more than 20° F. below the indoor control temperature of 75° F. the indoor air temperature dropped to below 63° F., producing an uncomfortable situation. This situation could have been corrected by reducing the window openings as in a normally occupied residence, but for test purposes all windows were fully open. The room-to-room temperature unbalance was increased approximately 1° F. during the night air cooling test series. This occurred because the west end of the residence, bedrooms and bath, which were already overcooled, were overcooled more because

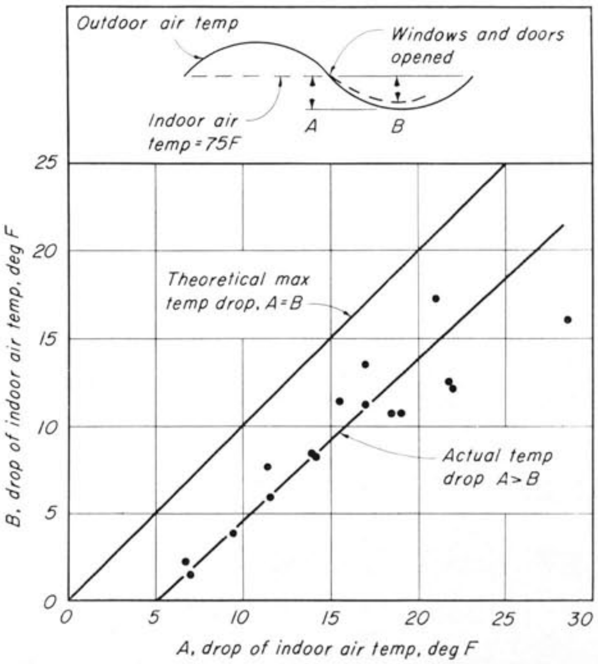


Figure 22. Comparison of indoor and outdoor air temperature drop during night-air cooling

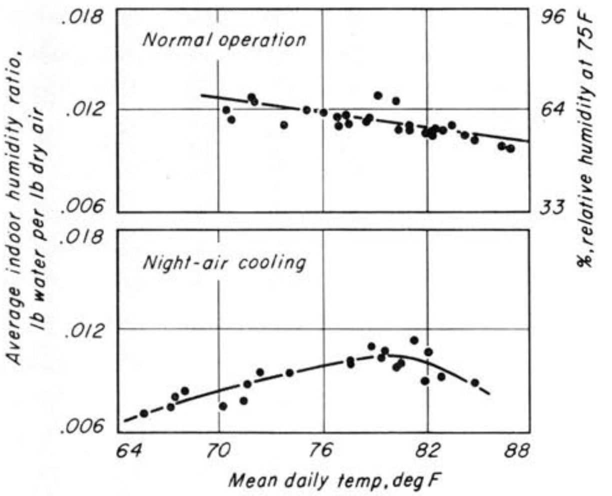


Figure 23. Comparison of indoor humidity ratios with and without night-air cooling

the west end of the structure warmed more slowly from solar radiation than the east end. The temperature readings upon which the balance data were based were obtained at 3:00 p.m. and the west end of the structure was cooler because of the cooling during the previous night. This problem could have been partly eliminated by proper adjustment of the balancing dampers.

The floor to ceiling level temperature differentials were reduced approximately 2° F. at design outdoor conditions as a result of the cooler surface temperatures.

The effect of night-air cooling on the daily average indoor humidity ratio is shown in Figure 23. The upper part of the figure is for normal operation and the lower part is for night-air cooling. The humidity ratio under normal operation de-

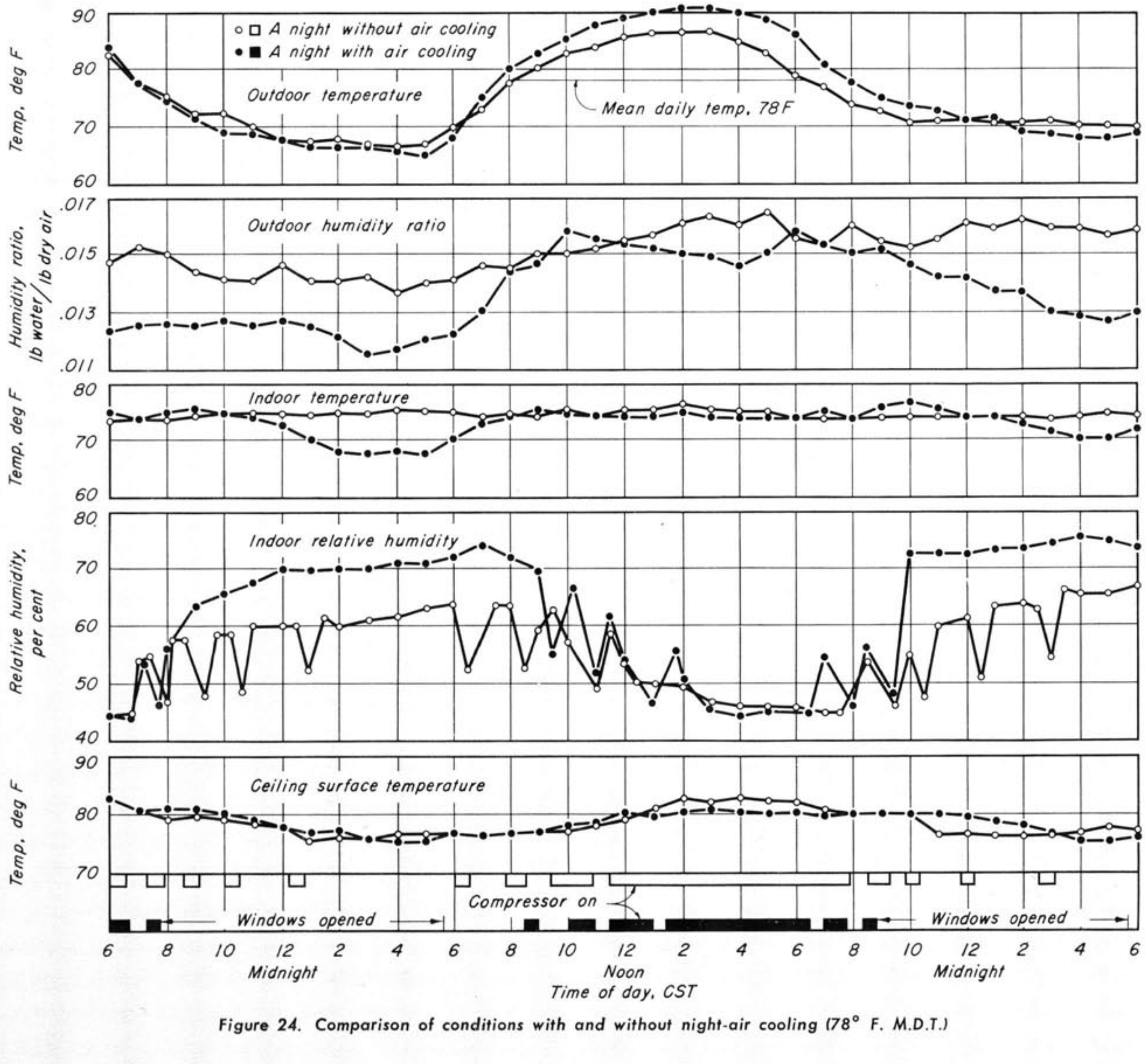


Figure 24. Comparison of conditions with and without night-air cooling (78° F. M.D.T.)

creased as the mean daily temperature increased. With night-air cooling the average indoor humidity ratio increased as the mean daily temperature increased until a mean daily temperature of 80° F. was attained. At mean daily temperatures above 80° F. the indoor humidity ratios were similar to those during normal operation because the windows and doors usually remained closed (this was true for 5 of 7 days above 80° F. M.D.T.) subject to the test procedure stated earlier. The average indoor relative humidity for normal operation is indicated on the right-hand ordinate, based on 75° F. dry-bulb temperature. A similar scale cannot be applied to night-air cooling because the indoor dry-bulb temperature was not constant.

A graphical comparison of the indoor and outdoor conditions on similar days with and without night-air cooling is shown in Figures 24 and 25. Figure 24 is a comparison of days with mean daily temperatures of approximately 78° F., near the upper limit at which night-air cooling could be utilized. Figure 25 is a comparison of days with mean daily temperatures of approximately 72° F.

On days of 78° F. M.D.T. a large increase in relative humidity and a small increase in indoor air temperature occurred after opening the windows and doors. For the day shown the relative humidity increased 25% and the indoor air temperature increased 1° F. A build-up of relative humidity also occurred during the evening hours when the windows were not open but the increase for the day shown was only 5%. With night-air cooling the drop in indoor air temperature was partially counteracted, as far as the sensation of

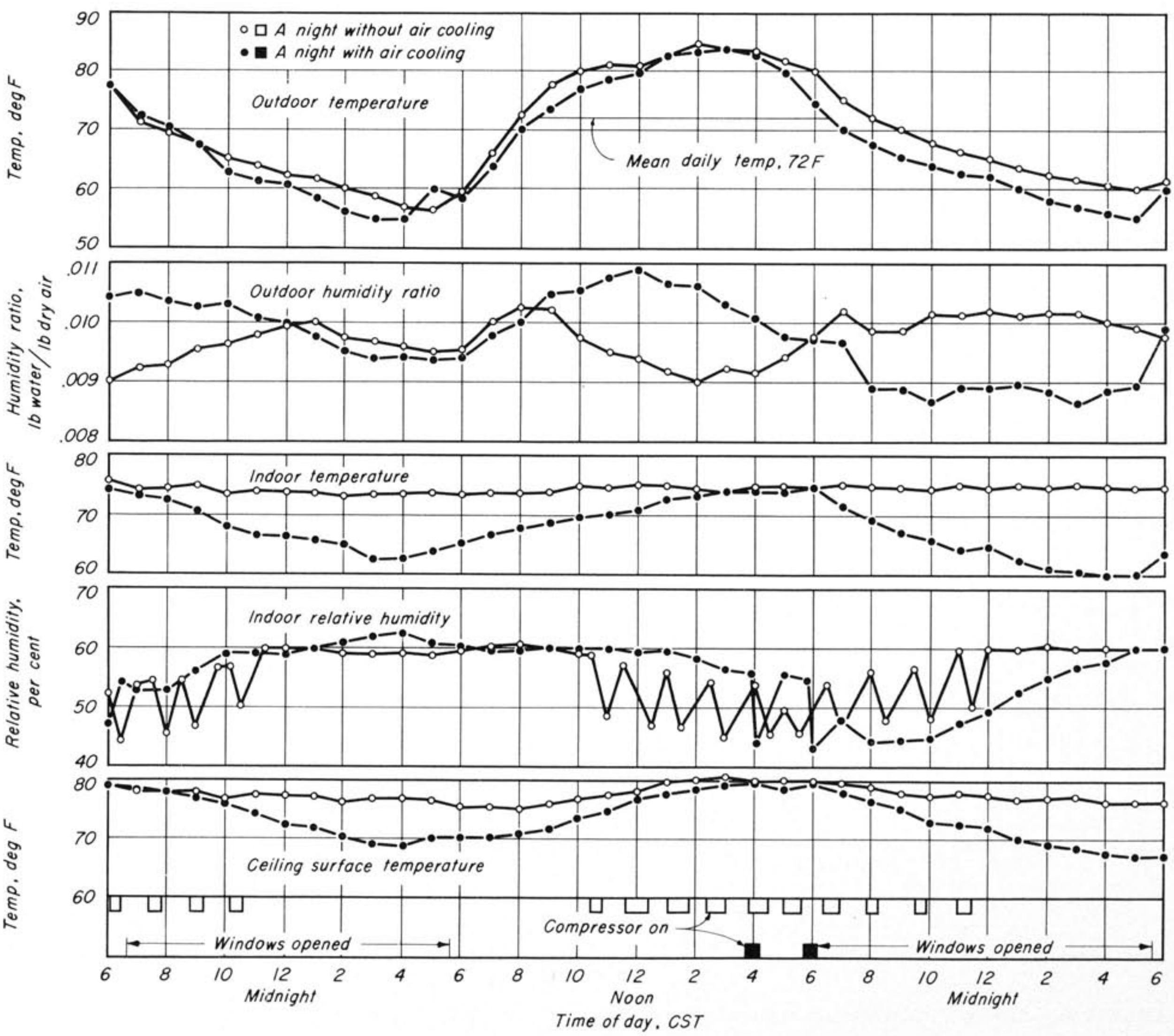


Figure 25. Comparison of conditions with and without night-air cooling (72° F. M.D.T.)

warmth was concerned, by the increase in relative humidity, although on some occasions the relative humidity was high enough to produce discomfort. The relative humidity with night-air cooling remained higher until after the first two or three compressor operations on the following day, and after this the indoor relative humidity was the same with or without night-air cooling until the windows and doors were again opened. A comparison of ceiling surface temperatures reveals that there was no significant difference in the subcooling of the structure during the night with or without night-air cooling at this mean daily temperature. Therefore, the reduction in operating time was primarily due to elimination of compressor cycles during the period when the windows and doors were open and not to subcooling of the structure.

On days of 72° F. mean daily temperature the relative humidity after opening the windows was not significantly different from the relative humidity under normal operating conditions with the windows closed. At 72° F. M.D.T. the indoor relative humidity was largely dependent upon the outdoor humidity ratio. For example, during the first night shown the indoor relative humidity was approximately the same with or without night-air cooling. During the second night shown the indoor relative humidity with night-air cooling was much lower than without night-air cooling. This was due to the lower outdoor humidity ratio that occurred when night-air cooling was utilized. The indoor air temperature decreased to 60° F. and 62° F. during the two nights shown. At 72° F. M.D.T. the reduction in compressor operating time was primarily due to subcooling of the structure as evidenced by the much cooler ceiling surface temperatures that occurred with night-air cooling.

From the viewpoint of comfort, as indicated by temperature and relative humidity, night-air cooling was most effective on days of mean daily temperature from 70° to 76° F. At mean daily temperatures above 76° F. the increase in relative humidity and temperature subsequent to opening the windows and doors was a source of discomfort. Below 70° F. M.D.T. the indoor temperature decreased to an uncomfortably low level in the early morning. This could have been corrected by only partially opening the windows, as mentioned above, but this would have reduced the savings in operating costs discussed in the following paragraph.

C. REDUCTION OF OPERATING COST

The reduction in compressor operation time as a result of night-air cooling is shown in Figure 26. The datum temperature (mean daily temperature above which compressor operation occurred) increased from 64° F. for normal operation to 69° F. with night-air cooling. For days of less than 80° F. M.D.T. the compressor operated approximately 4 hours less per day because of night-air cooling. Above 80° F. M.D.T. the windows and doors usually remained closed and, consequently, there was no difference in operation time.

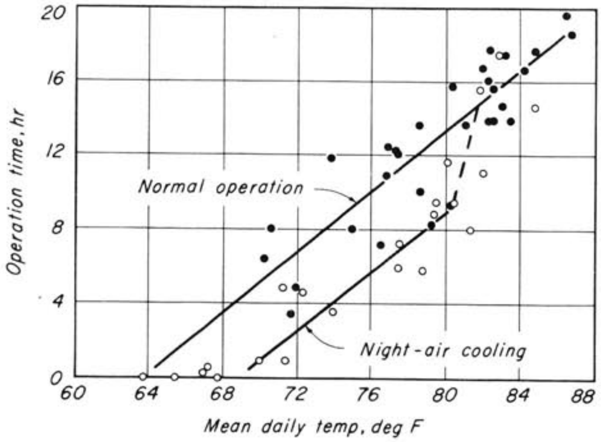


Figure 26. Compressor operating time

X. FORCED ATTIC VENTILATION TO REDUCE COOLING LOAD

A. OBJECTIVE OF INVESTIGATION

During the 1956 cooling season an investigation was conducted to determine the effectiveness of forced attic ventilation to reduce attic air temperatures and heat gain through the ceiling.

Heat gain through the ceiling is an important part of the total cooling load of a residence. In Research Residence No. 2 the heat gain through the ceiling represented 24% of the total calculated sensible heat gain under design conditions. Reduction of this large part of the heat gain may result in a reduction of the required installed capacity in many residences and will certainly result in some reduction in operating costs.

Air-cooled mechanical refrigeration units are sometimes located in attic spaces. The units may utilize attic air for condenser cooling and this may also reduce the attic air temperature. If the air flow rate is sufficient to reduce the attic air temperature to approximately outdoor air temperatures no increase in condensing temperature and consequent loss of capacity will result.

A system which utilizes ceiling diffusers and/or has the refrigeration unit located in the attic must also have ducts located in the attic space. Any reduction in attic air temperature reduces the heat gain to ducts and the operating costs and required capacity of the unit.

B. DESCRIPTION OF ATTIC

The residence had a black asphalt shingled gable-type roof extending east and west. The pitch of the roof was 5¾ vertical to 12 horizontal. The roof formed a 3-foot 10-inch overhang over the south wall and a 3-foot 4-inch overhang over the east section of the north wall. The roof may be seen in the photograph of the residence, Figure 1. A 1-inch width soffit vent covered with 16-mesh screen extended along the entire north and south eaves. There were gable vents at each end of the attic, also covered with 16-mesh screen. The attic ventilation fan was installed in the east gable. To prevent short-circuiting of the ventilation air the east gable vent was sealed. The gross area of the soffit vents was 6.25 square feet and the gross area of the west gable vent was 0.92 square feet for a total of 7.17 square feet. The free area of the 16-mesh screen was approximately 50% of the gross area. The total free area of the attic vents was, therefore, approximately 3.6 square feet. The total volume of the attic, including the space formed by the overhangs, was 4,684 cubic feet. The total exposed surface area of the room and gable ends including the sections of roof forming the overhangs was 1,663 square feet. The area of the ceiling was 1,040 square feet. The ceiling construction consisted of 8-inch open-web steel joists with 3/16-inch cement asbestos board panels bolted to the bottom of the joists to form the finished ceiling. Sufficient insulation was placed between joists to obtain an over-all coefficient of heat transmission (U) of 0.07 B.t.u. per hour per square foot.

The air conditioning duct system was the same perimeter system as used during the 1954 and 1955 cooling season investigations. A schematic diagram of the attic ventilation fan, including the controls, is shown in Figure 27. The fan was a 24-inch axial flow (propeller) type. It was belt

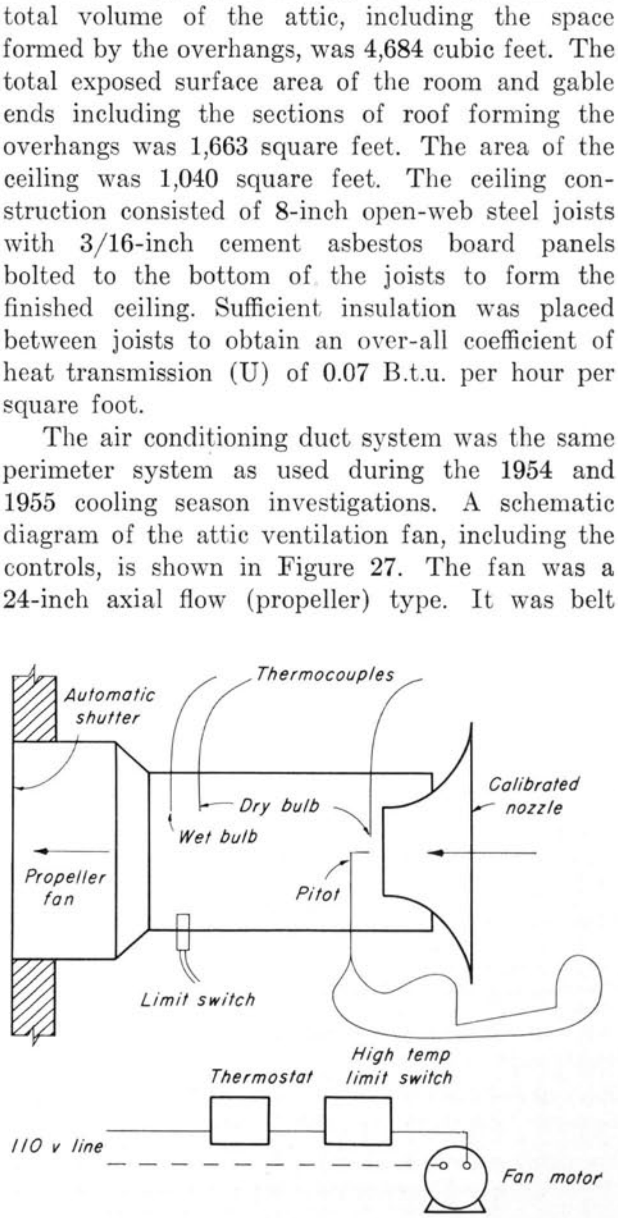


Figure 27. Schematic diagram of attic fan installation and wiring

driven by a ⅓-h.p. electric motor. The rated free air delivery with automatic shutters attached was 2,645, 3,130, 3,630, and 4,285 c.f.m. for fan speeds of 439, 492, 546, and 623 r.p.m., respectively. As shown in Figure 27b, the fan operation was controlled by an "off-on" thermostat located in the attic space. A manual resetting high-temperature limit switch was included in the control circuit to stop fan operation when the attic air temperature exceeded 165° F. The element of the limit switch was mounted in the fan inlet duct. The fan thermostat was set to cut on at 85° F. and off at 75° F. Because of poor calibration of the thermostat and/or lack of adequate radiation shielding of the thermocouples, the average "on" temperature indicated by an air thermocouple was approximately 89° F. The 10° F. differential between "on" and "off" temperature was maintained.

A calibrated nozzle and pitot tube were used to measure the attic ventilation rates. The nozzle was attached to a 20½-inch-diameter circular duct approximately 4 feet long. The circular duct was attached to the square fan housing by a sheet metal transition section.

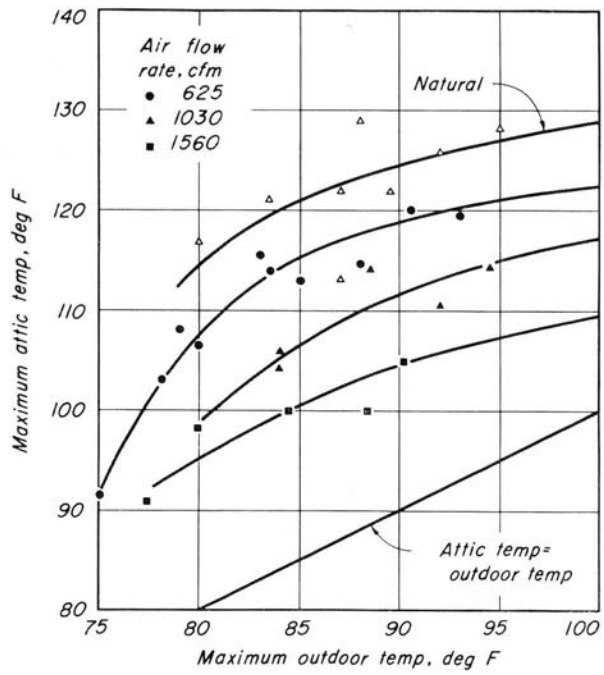


Figure 28. Relation between maximum attic air temperature and maximum outdoor air temperature at various attic ventilation rates

C. OPERATING CONDITIONS

Three test series were conducted during the investigation. The attic ventilation rates were 625, 1,030, and 1,560 c.f.m. standard air. Comparison is made between these three test series and data obtained during a previous cooling investigation with similar experimental conditions except that the attic ventilation was by natural convection.

The residence was occupied by a family of two adults and was cooled 24 hours per day. The indoor air control temperature was 75° F. Attic air temperatures were largely dependent upon the amount of solar radiation received by the attic surfaces. Since clear days are of most importance from a design viewpoint the data presented were restricted to relatively clear days.

D. ATTIC AIR TEMPERATURE

The attic air temperature at a point 2 feet below the south roof surface, 3 feet from the attic ridge line, and 13 feet from the west gable end (the end opposite the fan) was continuously recorded. Investigation indicated that the temperature at this location was representative of the temperatures at this level.

The relation between maximum attic air temperature and maximum outdoor air temperature for the three forced ventilation rates and natural ventilation is shown in Figure 28. The maximum intensity of solar radiation measured on a horizontal surface ranged from 367 to 256 B.t.u.h. per square foot for the days included. The lower intensities occurred during the test series with ventilation rates of 1,030 and 1,560 c.f.m. which was conducted during August. The difference in solar intensity was a result of change in solar declination. The maximum summer solar declination occurred June 21; therefore, the reduction in attic air temperature for the two larger ventilation rates was partly due to reduction of solar radiation received by the attic surfaces. The maximum attic air temperature increased rapidly with small increases in outdoor air temperature in the low outdoor air temperature range. In the high outdoor air temperature range the attic air temperature increased only slightly as the outdoor air temperature increased and approached a limiting temperature for each ventilation rate.

The maximum attic air temperature usually occurred 2 hours before the maximum outdoor air temperature and 2 hours after the maximum roof surface temperature. The over-all time lag of the roof-attic-ceiling structure as indicated by the dif-

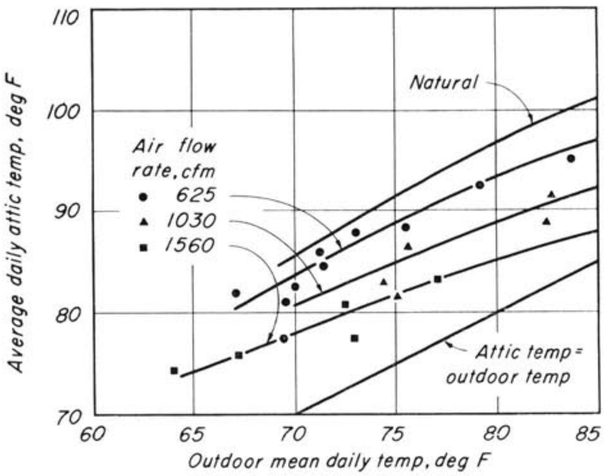


Figure 29. Relation between average attic air temperature and mean outdoor air temperature at various attic ventilation rates

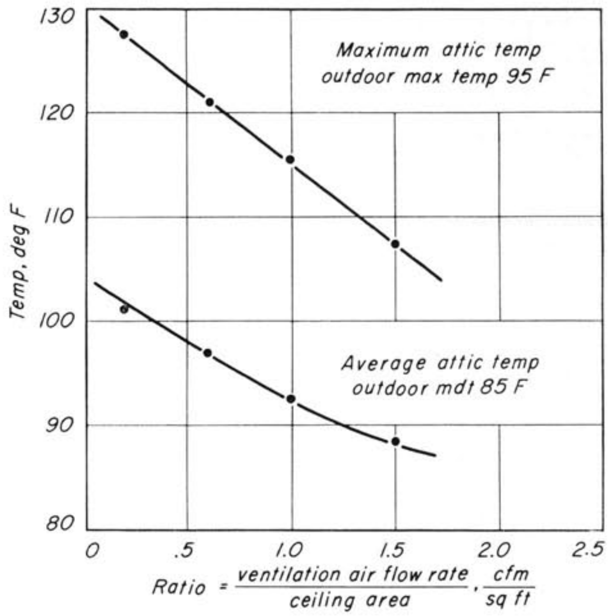


Figure 30. Effect of attic ventilation rate on maximum and daily average attic temperature on a design day

ference in times of occurrence of the maximum roof surface and maximum ceiling surface temperatures was 4.5 hours. Attic ventilation rate had no significant effect on the time lag.

The relation between average daily attic temperature and mean daily outdoor air temperature is shown in Figure 29 for the four ventilation rates. Increased attic ventilation produced a reduction in the average daily attic temperature. The reduction was largest at high mean daily temperatures.

The effect of attic ventilation rate on the maximum and daily average attic temperatures on a design day is shown in Figure 30. The temperatures were obtained from the curves of Figure 28 and Figure 29. The ventilation rates are expressed in cubic feet per minute per square foot of ceiling area. This is but one of three suggested possible methods of relating ventilation rate to size of house. The attic ventilation rate may also be expressed as number of air changes per hour or as cubic feet per minute per square foot of attic envelope area. A mathematical analysis revealed that the difference in total daily solar radiation received by the attic envelope of a house with various roofs from flat to a 6/12 pitch was less than 7.5%. Also, the direction in which the house faced caused a difference in the total solar radiation received of less than 2% as did the location of the house at either 30° or 40° latitude. Therefore, since the purpose of ventilation air is to remove the heat and since the solar heat to be removed is principally a function of ceiling area, cubic feet per minute per square foot of ceiling area provides a better index of attic ventilation. For example, consider a house of 1,000-square foot ceiling area. If this house had a flat roof with an average attic depth of 1 foot a ventilation rate of 1,000 c.f.m. would be equivalent to 60 air changes per hour. If the same house had a 5/12 pitch gable roof the attic volume would be 2,600 cubic feet, and 1,000 c.f.m. ventilation rate would be equivalent to 23 air changes per hour. The solar radiation incident upon the two roofs and, therefore, the heat transfer into the attic would be nearly equal in each instance, as would the resulting attic temperature.

The natural ventilation air-flow rate was calculated by Equation 4 in Chapter 11 of the *Heating, Ventilating, and Air Conditioning Guide.*[11] For an attic-outdoor temperature difference of 35° F. the natural ventilation rate was estimated to be 200 c.f.m. This ventilation rate was used to plot the minimum ventilation rate data in Figure 30 (also used in Figure 31).

As the attic ventilation rate was increased up to 1.5 c.f.m. per square foot of ceiling area the maximum attic temperature decreased in direct proportion. Extrapolation of the curve to a temperature equal to outdoor air temperature, 95° F., indicates that a ventilation rate of 2.26 c.f.m. per square foot would reduce the attic temperature on a design day to equal outdoor air temperature. However, as the maximum attic temperature approached 95° F. the cooling effect of the ventilation

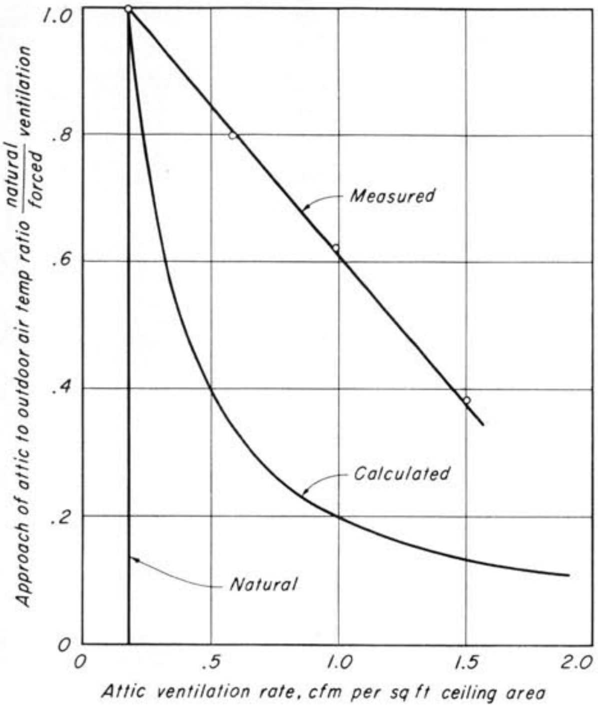


Figure 31. Approach of attic air temperature to outdoor air temperature as affected by attic ventilation rates

air would be reduced and the attic air temperature would only approach outdoor air temperature. The approach of maximum attic temperature to maximum outdoor air temperature has been calculated to determine the lowest maximum attic temperature that could be attained. The heat transfer through the roof, the heat storage in the attic, and the heat transmission through the ceiling were assumed constant. Using the natural ventilation study as a base, the ratios of approach of maximum attic air temperature to maximum outdoor air temperature with natural ventilation to the approach with forced ventilation are shown in Figure 31 as a function of ventilation rate in cubic feet per minute per square foot of ceiling area. The calculated data are compared with measured data obtained from Figure 28. The calculated curve in Figure 31 indicates that a small increase in the attic ventilation rate up to 1 c.f.m. per square foot will cause a large reduction in the maximum attic temperature. Increasing the ventilation above this value has little effect upon the maximum attic temperature, according to the calculated curve.

The measured approach of maximum attic air temperature to maximum outdoor air temperature was considerably larger than the calculated approach. This large difference was principally attributed to errors in the three assumptions made above in order to calculate the approach. The heat transfer through the roof would increase, the heat storage would decrease, and the ceiling heat transfer would decrease because of the reduced attic temperatures at higher attic ventilation rates. These three items would not remain constant as assumed. Table 16 lists the vertical temperature profile in the south attic at 3:00 p.m. during a day in each test series. From this table it may be seen that increasing the ventilation rate increased the difference in roof and underside roof surface temperatures, decreased the attic air temperature, and decreased the difference in topside and room side insulation temperature. The measured data curve would be expected to parallel the calculated curve at some ventilation rate larger than the 1.5 c.f.m. per square foot studied.

Table 16

Vertical Temperature Profile in South Attic at 3:00 p.m.

Ventilation Rate, c.f.m.	Natural 200[a]	625	1030	1560
Date	7-22-55	7-26-56	8-5-56	8-27-56
Outdoor Air Temperature, °F.	89.6	90.4	90.6	90.4
Solar Intensity,[b] B.t.u.h. ft.^{2}	278	225	221	172
Wind Velocity, m.p.h.	4	7	8	9
Wind Direction	SW	SW	W	SW
Temperature, °F.				
Roof Surface	142.1	139.8	137.7	135.2
Underside Roof Surface	132.6	128.4	123.6	119.6
Air, 2 Inches Below Roof	130.3	124.4	118.1	112.3
Air, 2 Feet Below Roof	125.0	117.1	110.0	103.4
Insulation, Topside	121.7	114.5	110.0	106.0
Insulation, Room Side	84.3	85.8	84.8	82.6
Ceiling Surface, South Bedroom	81.7	81.3	81.1	79.5

[a] Estimated.
[b] Measured on a horizontal surface.

The diminishing effect of increased attic ventilation on the daily average attic temperature for a design day (85° F. M.D.T.) is indicated in Figure 30. The curve indicates that increases in attic ventilation rates above 1.5 would result in only small reductions of daily average attic temperatures. From the viewpoint of temperature reduction a ventilation rate of approximately 2 c.f.m. per square foot of ceiling area would be the maximum effective ventilation rate in Research Residence No. 2.

E. CEILING HEAT GAIN

The reduction in ceiling heat gain on a design day (95° F. maximum outdoor air temperature, 85° F, outdoor mean daily temperature) resulting from forced attic ventilation is shown in Figure 32. The reduction is based on the ceiling heat flow with natural ventilation. Curves are shown for the

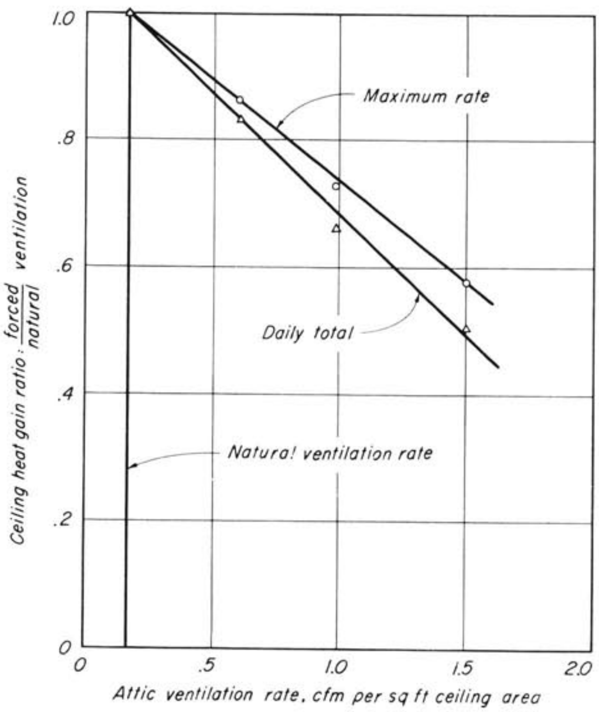


Figure 32. Reduction of ceiling heat gain by forced attic ventilation on a design day

reduction in both the maximum heat-gain rate and the total daily heat gain. With natural attic ventilation the maximum ceiling heat gain on a design day was 3,740 B.t.u.h. and the total daily heat gain was 47,800 B.t.u. The ceiling heat-gain rates were obtained by means of heat-flow meters installed between the ceiling surface and the insulation. Since the meters were placed between the ceiling joists, the measured rates were increased by the ratio of the calculated over-all heat-transmission coefficient of the ceiling (0.07 B.t.u.h. per square foot) to the calculated heat transmission of the spaces between joists (0.06 B.t.u.h. per square foot).

Since the calculated design heat gain through the ceiling constituted only 24% of the design sensible heat gain of the residence the reduction of 43% in maximum ceiling heat gain rate obtained with 1.5 c.f.m. per square foot attic ventilation rate was equivalent to a reduction of 10.3% in design sensible heat gain or 7.9% of the design total heat gain. In Research Residence No. 2, which was well insulated, the use of increased attic ventilation would not permit any reduction in capacity of the equipment installed. The effect of attic ventilation on the design gain in an uninsulated residence is discussed in a later section.

Forced attic ventilation reduced the daily ceiling heat gain by approximately 50%. The effect of the reduction in daily total heat gain on the operating costs is discussed in a later section.

F. ATTIC FAN OPERATION

An increase in the attic ventilation rate caused a decrease in the number of hours of fan operation per day, as shown in Figure 33, a plot of hours of attic fan operation per day versus mean daily temperature. The reduced operating time with increased ventilation rates was due to more rapid cooling of the attic space after sundown. Since the operation of the fan was controlled by a thermostat which started the fan when the attic temperature was approximately 89° F. the fan started at about the same time with all ventilation rates. The 79° F. cut-off temperature was reached earlier in the evening because of the increased "cooling capacity" of the attic fan at higher ventilation rates.

The fan cycled only once each day under all weather conditions with the three ventilation rates studied. However, a small increase of attic air temperature occurred immediately after the fan ceased operation because of heat storage in the attic structure. On days approaching design conditions the temperature increase was negligible with the low ventilation rate and was about 2° F. with the highest ventilation rate.

Studies were not conducted to determine the

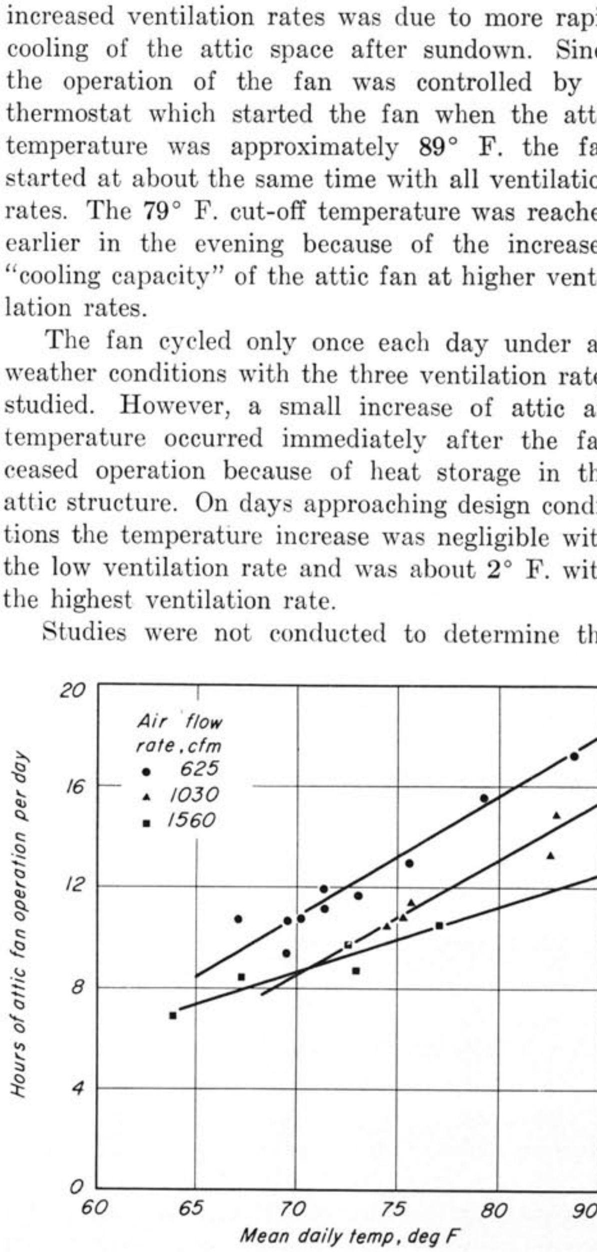


Figure 33. Attic fan operation time with various attic ventilation rates

Table 17

Net Reduction of Daily Operating Cost Due to Increased Attic Ventilation

Ventilation Rate, c.f.m.	Design Day, M.D.T. = 85° F. Reduction in Condensing Unit Operating Cost,[a] cents per day	Attic Fan Operating Cost,[b] cents per day	Net Saving in Operating Cost, cents per day	Average Day, M.D.T. = 75° F. Reduction in Condensing Unit Operating Cost, cents per day	Attic Fan Operating Cost, cents per day	Net Saving in Operating Cost, cents per day
Natural	0	0	0	0	0	0
625	5.3	2.8	2.5	3.9	2.0	1.9
1030	11.2	2.9	8.3	8.3	2.0	6.3
1560	16.3	3.9	12.4	12.1	3.1	9.0

[a] Based on operating cost of 12.6 cents per hour for water-cooled condensing unit in Research Residence No. 2. Power cost = 2.5 cents per kw.h. and water cost = 50 cents per 1,000 gallons.

[b] Based on measured fan operating time and manufacturers' data for fans having free air delivery at least equal to test delivery. Motor efficiency assumed 50%. Power cost = 2.5 cents per kw.h.

effect of control temperature on the fan operation or effectiveness of ventilation. However, previous studies conducted on a design day in which the attic ventilation was natural indicate that the attic and outdoor air temperatures increased at the same rate and were equal during the early morning. At an outdoor temperature of approximately 85° F. the attic air temperature began to increase faster than the outdoor air temperature. Therefore, in Research Residence No. 2, beginning attic fan operation at temperatures below 85° F. would not have increased the effectiveness of forced attic ventilation on design days. Some small advantage could possibly have resulted from lower cut-on temperatures on milder days.

Lower cut-on and cut-off temperatures or a larger thermostat differential would have increased the attic fan operation time. Higher cut-on and cut-off temperatures or a smaller thermostat differential would have had the opposite effect.

With no wind the outdoor-attic pressure differentials were 0.002, 0.005, and 0.013 inch of water with attic ventilation rates of 625, 1,030, and 1,560 c.f.m., respectively. The pressure differentials, which were a measure of the pressure loss of the attic vents, were equivalent to a pressure loss of one velocity head based on the free area of the vents.

G. OPERATING COST REDUCTION

Because of the resistance imposed on the fan by the equipment used to measure the ventilation rates (Figure 27) the measured electrical inputs to the installed fan could not be used for comparison of operating costs. The measured outdoor-attic pressure differentials indicated that a fan installed in Research Residence No. 2 would operate at approximately free-air conditions. Representative fans that had free-air capacities equal to or slightly larger than the test ventilation rates were selected from manufacturer's data to determine the fan operating cost. The average motor sizes listed for suitable fans were 1/24, 1/20, and 1/12 h.p. for ventilation rates of 625, 1,030, 1,560 c.f.m., respectively. The corresponding electrical inputs, assuming a motor efficiency of 50%, were 62, 75, and 125 watts. The cost of operating the fan on a design day and on an average day is shown in Table 17 for each ventilation rate. The fan operation time for each condition was obtained from Figure 33. The costs are based on an electrical power rate of 2.5 cents per kw.h.

The reductions in condensing unit operating costs listed in Table 17 were determined from the reduction in daily ceiling heat gain used to plot the daily total curve in Figure 32 for a design day and from similar data for an average day. The reduction in condensing unit operation time was based on the assumption that a reduction of 1 hour of condensing unit operation would result from each 18,000 B.t.u. reduction in the sensible heat gain because the sensible heat capacity of the water-cooled unit was 18,000 B.t.u.h. The cost per hour of condensing unit operation was based on measured electrical and water requirements of the condensing unit (2.59 kw. and 122 g.p.h.), an electrical power rate of 2.5 cents per kw.h. and a water rate of 50 cents per 1,000 gallons. Lower electrical or water rates or the use of a water conserving device would reduce the net savings due to forced attic ventilation. The reduction in operating time could not be obtained from plots of measured time of compressor operation as a function of mean daily temperature because of variations in the daily load caused primarily by occupancy.

The net savings due to forced attic ventilation increased as the ventilation rate increased. A limiting ventilation rate above 1.5 c.f.m. per square foot of ceiling area would exist where further increases in ventilation would reduce the net savings, as indicated by Figures 31 and 32 and the previous discussion.

H. EFFECT OF ATTIC VENTILATION ON COMFORT

Attic ventilation had a small effect on the comfort conditions of the conditioned space. Increased attic ventilation reduced the ceiling surface temperatures and consequently reduced the mean radiant temperature (M.R.T.) of the conditioned space. The living room ceiling surface temperatures measured at 3:00 p.m. in the living room are shown in Figure 34 as a function of indoor-outdoor temperature difference and ventilation rates studied. The low ceiling surface temperatures measured during the test series with the maximum ventilation rate were partly due to the larger daily outdoor temperature range during this test series.

Because the space mean radiant temperature is difficult to determine and varies between locations, another similar index of the radiant heat transfer, average surface temperature (A.S.T.), may be used to compare the effect of attic ventilation rates. The ceiling area of the living room was approximately one-fourth of the total surface area of the room, therefore a 4° F. reduction in ceiling surface temperature resulted in a 1° F. reduction in average surface temperature. The reduction in average surface temperature at design conditions (20° F. indoor-outdoor temperature difference) below the average surface temperature with natural ventilation was 0.4° F., 0.6° F., and 1.1° F. with ventilation rates of 625, 1,030, and 1,560 c.f.m., respectively. Similar reductions in the ceiling surface temperatures and average surface temperatures were observed in the other rooms.

A reduction in the room air temperature differentials would be anticipated from the reduction of the ceiling surface temperatures at increased ventilation rates. However, the room air temperature differentials in the living zone (4-inch to 60-inch level) were not significantly reduced and only a small reduction in the floor-to-ceiling level temperature differentials resulted from increased attic ventilation.

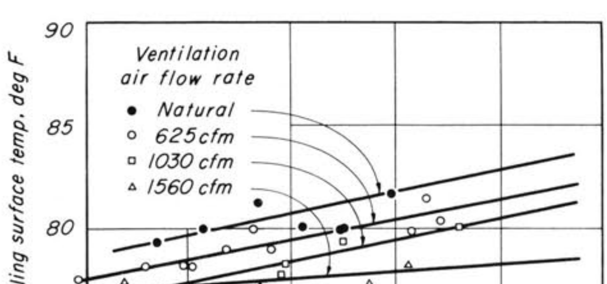


Figure 34. Ceiling surface temperature in the living room at various attic ventilation rates

I. ATTIC-AIR DEW POINT

Dry-bulb and wet-bulb thermocouples located in the fan inlet duct indicated that the attic-air dew point was the same as the outdoor-air dew point, regardless of the ventilation rate. This is significant in that outdoor-air conditions can be used to determine whether condensation will occur on ducts located in ventilated attics. The attic-air dew point varied from 55.1° F. to 78.6° F. at the time of maximum load (3:00 p.m.). The attic-air relative humidity varied from 27% to 71% with an average of approximately 40%. This does not represent the daily maximum relative humidity, which occurs at night. However, the maximum relative humidity, since it occurs at night, occurs when there is little or no cooling being performed.

J. APPLICATION OF RESULTS

Because the ceiling of Research Residence No. 2 was well insulated the reduction in the design maximum ceiling heat-gain rate resulting from forced attic ventilation amounted to only 1,590 B.t.u.h. with the largest ventilation rate investigated. The reduction was only 7.9% of the total calculated design heat gain of the residence. To determine the reduction that might be obtained in

Table 18

Calculated Reduction of Ceiling Heat Gain at Design Conditions in Research Residence No. 2 for Various Combinations of Insulation and Ventilation

Ceiling Insulation, in.	Ventilation Rate, c.f.m./ft.^{2}	Design Ceiling Heat Gain,[a] B.t.u.h.	Percentage Reduction in Ceiling Heat Gain Due to: Insulation	Ventilation	Total
0	Natural	14,660	0	0	0
0	0.60	12,600	0	14.0	14.0
0	0.99	10,860	0	25.8	25.8
0	1.50	8,400	0	42.7	42.7
2	Natural	5,512	62.4	0	62.4
2	0.60	4,750	62.4	5.2	67.6
2	0.99	4,100	62.4	9.6	72.0
2	1.50	3,160	62.4	16.0	78.4
3⅝	Natural	4,160	71.6	...	71.6
3⅝	0.60	3,580	71.6	4.0	75.6
3⅝	0.99	3,090	71.6	7.3	78.9
3⅝	1.50	2,380	71.6	12.1	83.7

[a] Based on heat gain factors *Manual 11*,[7] reduction due to ventilation based on percentages obtained during investigation, ceiling area of 1.040 square feet.

houses with less ceiling insulation and to compare the relative effects of insulation and ventilation the maximum ceiling heat gain was calculated for a comparable house (ceiling area = 1,040 square feet) with zero, 2, or 3⅝ inches of ceiling insulation. The heat gain factors were obtained from *Manual 11.*[7] The percentage reductions in maximum heat gain determined in this investigation were assumed to apply also to houses with less insulation. The results of these calculations are listed in Table 18. The percentage reduction in ceiling heat gain is based on the heat gain of the ceiling with no insulation and only natural ventilation. The reduction due to insulation and the reduction due to ventilation are listed separately in order to compare the relative merits of each method of reducing the heat gain through the ceiling.

An analysis of the data in Table 18 reveals that the addition of 2-inch thick insulation is considerably more effective in reducing ceiling heat gain than the maximum ventilation rate. When 3⅝-inch thick insulation is added the effect of ventilation is almost insignificant. If 6-inch thick insulation were added to the ceiling the effect of ventilation would be even less.

XI. APPENDIX: CALCULATED TEMPERATURE RISE IN CIRCULAR DUCTS

A. OBJECTIVE

The objective of this mathematical investigation was to determine the effects of duct air temperature and velocity, ambient-air temperature, and combinations of insulation and vapor barriers on the heat gain to, and consequent temperature rise in, circular ducts utilized for cooling.

B. PROCEDURE

1. Uninsulated Ducts

The heat transfer from the surroundings to the conditioned air flowing in a duct depends upon the heat transfer coefficients of free convection and radiation at the outside surface of the duct and the coefficient of heat transfer of forced convection at the inside surface of the duct. The resistance to heat flow of the duct wall has been neglected because of the small thickness of the wall and the relatively large thermal conductivity of the sheet metal used in the manufacture of ducts. The heat transfer at the inside and outside surfaces must be equal when steady-state heat transfer exists. The heat balance can be expressed by the following equation:

$$q = h_i\, A(t_W - t_M) = (h_r + h_c)\, A\, (t_A - t_W) \quad \textbf{(A1)}$$

where q = heat transfer, B.t.u.h.

h_i = inside surface coefficient of forced convection heat transfer, $\frac{\text{Btuh}}{\text{ft}^2\ {}^\circ\text{F}}$

t_W = duct surface temperature, °F.

t_M = mean temperature of conditioned air, °F.

h_c = outside surface coefficient of natural convection heat transfer, $\frac{\text{Btuh}}{\text{ft}^2\ {}^\circ\text{F}}$

h_r = outside surface coefficient of radiant heat transfer, $\frac{\text{Btuh}}{\text{ft}^2\ {}^\circ\text{F}}$

A = area of duct surface, assumed equal for inside and outside surface because of small wall thickness, ft.²

t_A = ambient temperature, °F.

The temperature change of the air flowing in the duct is not a linear function of duct length, but for short sections of duct the change may be considered linear. It was assumed that the temperature change would be approximately linear in a 10-foot length; therefore, the area referred to is the wall area of 10 linear feet of duct.

The solution of Equation A1 requires an evaluation of the heat transfer coefficients for assumed ambient and duct air temperatures. The inside surface coefficient of heat transfer by forced convection was evaluated by the equation for forced convection in turbulent flow in horizontal pipes found in most heat transfer texts:[20]

$$h_i = 0.023\, \frac{k}{D} \left[\frac{VD\rho}{\mu}\right]^{0.8} \left[\frac{c\mu}{k}\right]^{0.4} \quad \textbf{(A2)}$$

where k = thermal conductivity of the fluid (air), $\frac{\text{Btuh}}{\text{ft}^2\ {}^\circ\text{F}}$

D = diameter of duct, ft.

V = average velocity of fluid (air), ft/sec

ρ = fluid density, $\frac{\text{lb}}{\text{ft}^3}$, standard conditions

μ = dynamic viscosity of fluid, $\frac{\text{lb}}{\text{sec ft}}$

c = specific heat of fluid, assumed constant, $0.24\ \frac{\text{Btu}}{\text{lb}\ {}^\circ\text{F}}$

$\frac{c\mu}{k}$ = Prandtl number

$\frac{VD\rho}{\mu}$ = Reynold's number

The properties of air were evaluated at the mean duct air temperature. The values of thermal conductivity were obtained from *Gas Tables*[21] by Keenan and Kaye. Over the range of temperatures from 32° to 122° F., the Prandtl number varied from 0.712 to 0.701 and an average value of 0.706 was used throughout the calculations. The inside surface coefficients of heat transfer were evaluated for duct air velocities from 400 to 1,200 f.p.m. in increments of 200 f.p.m. and mean duct

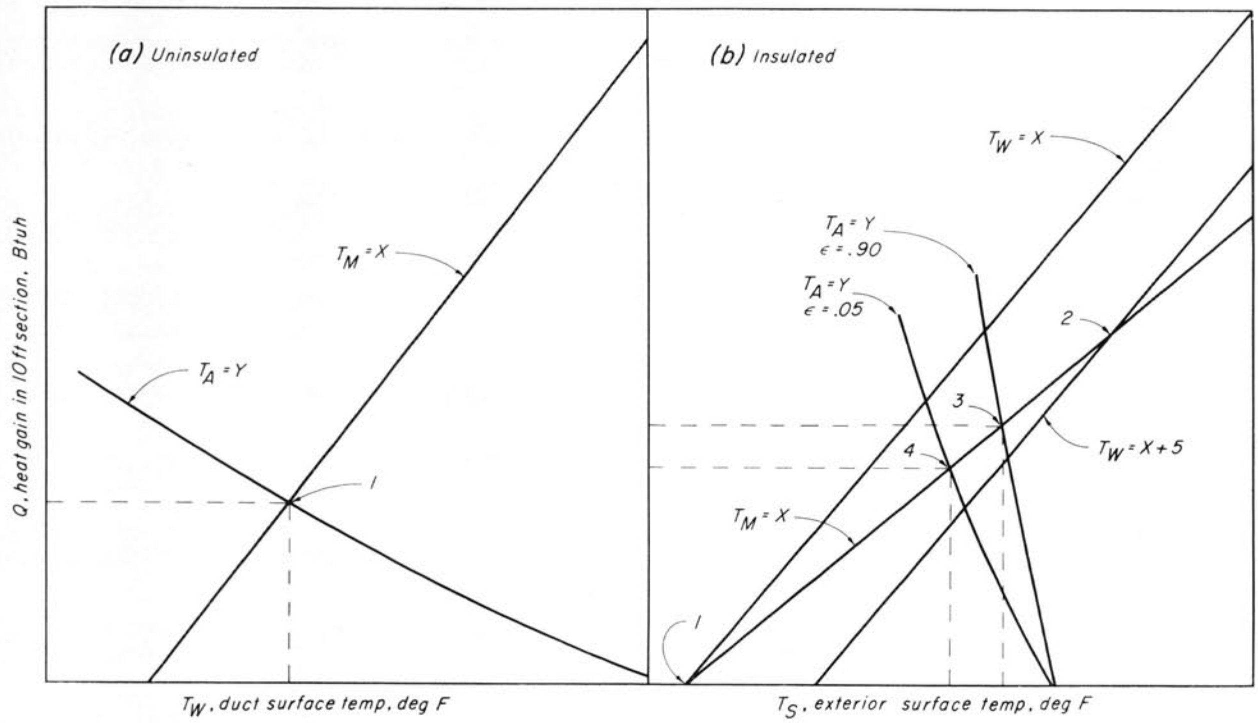


Figure 35. Graphical solution of equation for heat transfer to air flowing in ducts

air temperatures of 50° to 70° F. in increments of 5° F. for 4-, 5-, 6-, 7-, and 8-inch-diameter ducts.

The outside surface coefficient of heat transfer due to natural convection was determined by the equation found in most heat transfer texts:[20]

$$h_c = 0.53 \frac{k}{D} \left[\frac{D^3 \beta^2 \rho g \, (t_A - t_W)}{\mu^2} \right]^{0.25} \left[\frac{c\mu}{k} \right] 0.25 \quad \textbf{(A3)}$$

where β = temperature function, reciprocal of absolute temperature, 1/(degrees Rankine)

$$g = \text{gravitational constant, } 32.16 \frac{\text{ft lb}_f}{\text{lb}_m \text{sec}^2}$$

For the determination of the coefficient of natural convection the film temperature (the temperature of the thin stagnant layer of air adjacent to the surface) was assumed equal to the average of the ambient air and duct surface temperatures, and the properties of the air film, thermal conductivity, density, dynamic viscosity, and the temperature function were evaluated at the film temperature. The coefficients were determined for ambient temperatures of 75°, 100°, 125°, and 150° F. and duct surface temperatures of 55° to 100° F. in 5° F. increments.

The outside surface coefficient of radiant heat transfer was evaluated by the Stefan-Boltzmann[20] equation divided by the difference in ambient and duct surface temperatures, assuming a configuration factor of unity for a small duct located in a relatively large enclosure:

$$h_r = \delta\epsilon \frac{(T_A^4 - T_W^4)}{(T_A - T_W)} \quad \textbf{(A4)}$$

where δ = Stefan-Boltzmann constant, 0.173 x $10^{-8} \frac{\text{Btuh}}{\text{ft}^2 \ ^\circ\text{F}^4}$

ϵ = surface emissivity of duct

T_A, T_W = ambient temperature and duct surface temperature, degrees Rankine

Equations A1 through A4 apply to any thin wall duct. For the purpose of this analysis only gray oxidized zinc surfaces with a surface emissivity $\epsilon = 0.25$ were considered. The ducts installed in the residence were also of galvanized iron.

Radiation coefficients were determined for the same range of ambient and duct surface temperatures as the natural convection coefficients.

After the coefficients of heat transfer were determined the graphical method illustrated in Figure 35a was used to solve Equation A1 for the heat transfer rate and duct surface temperature.

In Figure 35a each side of Equation A1 is plotted as a function of the duct surface temperature, t_W. The left-hand side of Equation A1 represents the heat transfer through the inside surface. It is represented by a line designated $t_M = x$. For a constant duct air temperature the heat transfer increases linearly with duct surface temperature and is zero for $t_M = t_W$. The heat transfer to the outside surface of the duct at a constant ambient temperature $t_A = y$ is shown. As the duct surface temperature, t_W, increases, the temperature difference decreases and, therefore, the heat transfer would decrease to zero at $t_A = t_W$.

For a specified condition of duct air velocity, duct air temperature, and ambient temperature the intersection (1) of the two curves in Figure 35a provides a solution of Equation A1. The duct surface temperature and the heat transfer rate are obtained directly from the coordinates.

The temperature rise in a 10-foot length of duct was determined from the equation:

$$\Delta t = \frac{q}{60\, VA'\rho c} \qquad \textbf{(A5)}$$

where Δt = temperature rise of air, $\frac{°F}{10 \text{ lineal ft}}$

60 = factor to convert velocity from ft/min to ft/hr

A' = cross-sectional area of duct, ft^2

The duct surface temperature determines whether condensation will occur. Condensation of vapor will occur if the duct surface temperature is below the dew point of the ambient air. The dew point is a function of the ambient air dry-bulb temperature and relative humidity.

The mean duct air temperature, t_M, represents the temperature at the midlength of a 10-foot section of duct. The entering air temperature is obtained by subtracting one-half Δt from the mean temperature:

$$t_i = t_M - \frac{\Delta t}{2} \qquad \textbf{(A6)}$$

where t_i = duct air temperature at the entrance of the 10-foot section. The temperature rise was plotted as a function of entering temperature for the various duct air velocities and ambient temperatures. These graphs are discussed in a later section.

2. Insulated Ducts

The analysis for insulated ducts is similar to the analysis for uninsulated ducts except that the resistance to heat flow imposed by the insulation must be considered.

The heat transfer through the inside surface of the duct due to forced convection is the same as for uninsulated ducts. The heat transfer to the outer surface by natural convection and radiation is the same as for uninsulated ducts except that the outside surface temperature, t_S, is substituted for the duct surface temperature, t_W, and the outside surface area, A_S, is substituted for the duct surface area, A.

The heat transfer through the insulation is given by the conduction equation:

$$q = CA_M\,(t_S - t_W) \qquad \textbf{(A7)}$$

where C = thermal conductance of the insulation, $\frac{\text{Btuh}}{ft^2\ °F}$

A_S = log mean area of insulation, $\dfrac{A_S - A_W}{\ln \dfrac{A_S}{A_W}}$, ft^2

Setting the heat transfer through each resistance equal:

$$\begin{aligned} q &= h_i\, A_W\,(t_W - t_M) = C\, A_M\,(t_S - t_W) \\ &= (h_c + h_r)\, A_s\,(t_A - t_S) \end{aligned} \qquad \textbf{(A8)}$$

The inside surface coefficient of heat transfer by forced convection and the heat transfer rate were the same as evaluated for uninsulated ducts.

The coefficient of natural convection was calculated by a procedure identical with that for the coefficient for uninsulated ducts except that the diameter used in Equation A3 was the outer diameter of the insulation. The surface temperature was t_S instead of t_W.

The radiation coefficient was also calculated similarly to the coefficient for uninsulated ducts. Calculations were made for two types of vapor barriers, aluminum foil and duplex paper. An emissivity of 0.05 was assumed for aluminum foil and an emissivity of 0.90 was assumed for duplex paper, which consisted of two layers of kraft paper bonded together with a thin layer of asphalt. The conductivity of the insulation was assumed to be 0.27 $\frac{\text{Btuh}}{ft^2\ °F}$ per inch of thickness, which gave conductances of 0.27 $\frac{\text{Btuh}}{ft^2\ °F}$ for 1-inch thick insulation and 0.135 $\frac{\text{Btuh}}{ft^2\ °F}$ for 2-inch thick insulation. Calculations of the heat transfer rate through the

Table 19

Range of Variables in Analysis of Heat Gain to Ducts

Variable	Range
Duct Diameter, in.	4, 5, 6, 7, 8
Duct Air Temperature, °F.	50, 55, 60, 65, 70
Duct Air Velocity, f.p.m.	400, 600, 800, 1000, 1200
Ambient Temperature, °F.	75, 100, 125, 150
Insulation Thickness, in.	0, 1, 2
Vapor Barrier Emissivity	0.25 for uninsulated 0.05 and 0.90 for insulated

insulation were made for both thicknesses of insulation for duct surface temperatures from 55° to 75° F. in 5° F. increments and outside surface temperatures from 55° to 150° F. in increments of 5° F.

The solution of the heat transfer Equation A8 was solved graphically as indicated in Figure 35b. In this case the heat transfer was plotted as a function of the outside surface temperature, t_S.

The equation of heat transfer through the insulation yields lines of constant duct wall temperature represented by the lines $t_W = x$ and $t_W = x + 5$. The heat transfer, q, is zero when $t_W = t_S$ and increases linearly as t_S increases, for a constant duct surface temperature. For a constant outside surface temperature the heat-transfer rate decreases as the duct surface temperature increases, therefore, the line for $t_W = x + 5$ is below the line for $t_W = x$.

The heat transfer to the outer insulation surface is represented by the two lines designated $t_A = y$. The upper line is for the aluminum foil vapor barrier (emissivity = 0.05) and the lower line is for the duplex paper vapor barrier (emissivity = 0.90). The effectiveness of the lower emissivity vapor barrier in reducing the heat transfer rate and temperature rise is obvious.

The heat transfer through the inside film is dependent upon the mean duct air temperature and the duct surface temperature. If $t_M = t_W = x$, for example, the heat transfer rate is zero as represented by intersection (1) on Figure 35b. If $t_W = x + 5$ and $t_M = x$ the temperature potential is 5° F. and the heat transfer may be calculated for this condition and the value is represented by intersection (2) located on the $t_W = x + 5$ line. The intersection of the line of constant mean duct air temperature, t_M, with the line of constant ambient temperature, t_A, for either surface emissivity, intersection (3) or (4), provides a solution of Equation A8. The heat transfer rate and the outer surface temperature are obtained directly from the coordinates. The temperature rise and the entering temperature were determined as for uninsulated ducts and the temperature rise was plotted as a function of entering air temperature for the various combinations. These graphs are discussed in the following section.

C. RESULTS

1. General

Curves of temperature rise as related to entering temperature were plotted for all combinations of duct diameter, insulation thickness, duct air velocity, vapor barrier emissivity, and ambient temperatures. For convenient reference the ranges of these items are listed in Table 19.

2. Temperature Rise Curves

Temperature rise as related to entering temperature for 800 f.p.m. velocity and 75° and 125° F. ambient temperatures are presented in Figure 36 and Tables 20 and 21 for all duct diameters. Figure 36 is for uninsulated ducts at 75° and 125° F. ambient temperature. Table 20 is for 75° F. ambient air and ducts insulated with 1- and 2-inch thicknesses of insulation with vapor barrier surface emissivities of 0.05 and 0.90. Table 21 is similar to Table 20 but is for 125° F. ambient air. The temperature rise data obtained from the figure and tables show temperature rise per lineal foot in a 10-foot section. If the temperature rise in a duct longer than 10 feet is to be determined the duct must be divided into 10-foot segments. The following example illustrated in Figure 36 shows the procedure.

Table 20

Temperature Rise in 10-Foot Sections of Insulated Ducts, °F./ft. 75° F. Ambient Temperature

Entering Air Temperature, °F.	4-Inch Diameter				5-Inch Diameter				6-Inch Diameter				7-Inch Diameter				8-Inch Diameter			
Vapor Barrier Surface Emissivity	0.05		0.90		0.05		0.90		0.05		0.90		0.05		0.90		0.05		0.90	
Insulation Thickness, in.	1	2	1	2	1	2	1	2	1	2	1	2	1	2	1	2	1	2	1	2
50	.07	.05	.09	.06	.06	.04	.07	.04	.05	.03	.05	.03	.04	.03	.05	.03	.03	.02	.04	.02
55	.06	.04	.07	.05	.04	.03	.05	.04	.04	.02	.04	.03	.03	.02	.04	.02	.03	.02	.03	.02
60	.05	.03	.05	.03	.03	.02	.04	.03	.03	.02	.03	.02	.02	.02	.03	.02	.02	.01	.02	.01
65	.03	.02	.04	.02	.02	.02	.03	.02	.02	.01	.02	.01	.02	.01	.02	.01	.01	.01	.02	.01
70	.02	.01	.02	.01	.01	.01	.02	.01	.01	.01	.01	.01	.01	.01	.01	.01	.01	.01	.01	.01

Table 21

Temperature Rise in 10-Foot Sections of Insulated Ducts, °F./ft. 125° F. Ambient Temperature

Entering Air Temperature, °F.	4-Inch Diameter				5-Inch Diameter				6-Inch Diameter				7-Inch Diameter				8-Inch Diameter			
Emissivity	0.05		0.90		0.05		0.90		0.05		0.90		0.05		0.90		0.05		0.90	
Insulation Thickness, in.	1	2	1	2	1	2	1	2	1	2	1	2	1	2	1	2	1	2	1	2
50	.23	.16	.26	.17	.18	.12	.21	.13	.14	.10	.17	.10	.12	.08	.14	.09	.10	.07	.12	.07
55	.22	.15	.25	.16	.17	.11	.19	.12	.13	.09	.16	.10	.11	.07	.13	.08	.10	.06	.11	.07
60	.21	.14	.24	.15	.16	.11	.18	.12	.13	.08	.15	.09	.11	.07	.12	.08	.09	.06	.11	.07
65	.19	.13	.22	.14	.15	.10	.17	.11	.12	.08	.14	.09	.10	.06	.12	.07	.08	.05	.10	.06
70	.18	.12	.21	.13	.13	.09	.16	.10	.11	.07	.13	.08	.09	.06	.11	.07	.08	.05	.09	.06

Example 1. A 4-inch-diameter uninsulated duct is located in a basement where the ambient temperature is 75° F. The duct is 25 feet long. The air temperature at the entrance is 55° F. and the air velocity is 800 f.p.m. Determine the over-all air temperature rise and the exit temperature. Solution: With the entering air temperature of 55° F., Figure 36 indicates a temperature rise of 0.19° F/ft in the first 10-foot section. Therefore, the air temperature leaving the first 10-foot section is 55.0° F. plus 10 (0.19)° F. or 56.9° F. This is also the entering air temperature for the second 10-foot section. From Figure 36 the temperature rise in the second 10-foot section is 0.17° F/ft which results in a leaving temperature of 56.9° F. plus 10 (0.17)° F. or 58.6° F. The entering air temperature for the last 5 feet of duct is also 58.6° F. and the temperature rise in this section is 0.15° F/ft. The final temperature is 58.6° F. plus 5 (0.15)° F. or 59.4° F. The over-all air temperature rise is 59.4° F. minus 55.0° F. or 4.4.° F. Tables 20 and 21 permit the determination of temperature rise information for air flowing in insulated ducts with two vapor barrier surface emissivities and exposed to two ambient air temperatures. The temperature rise for unlisted entering air temperatures between 55° and 70° F. may be determined by interpolation.

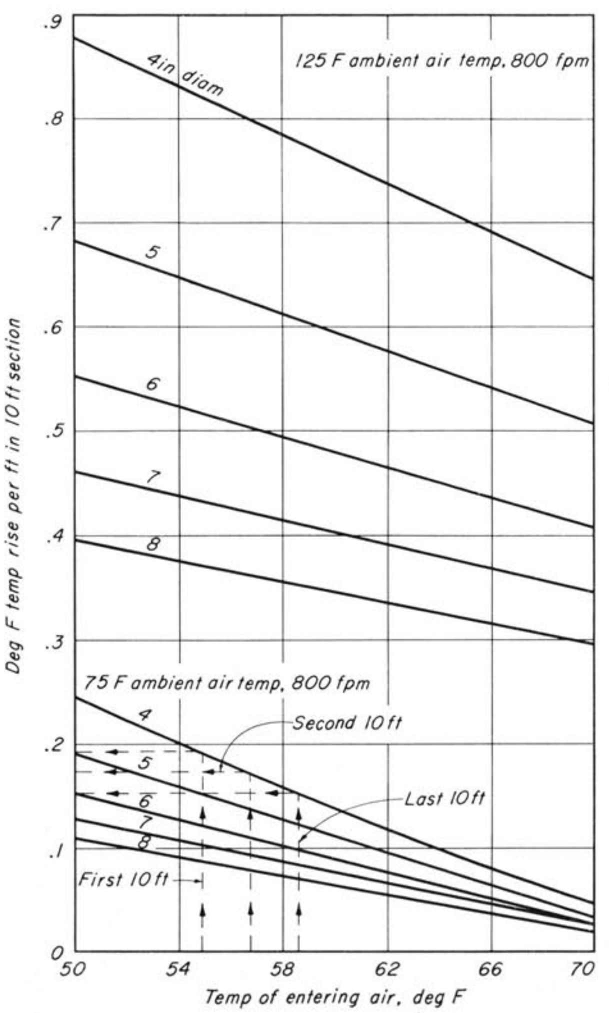


Figure 36. Temperature rise in uninsulated ducts

3. Velocity Correction Factors

The velocity correction factors, based on a velocity of 800 f.p.m. are presented in Table 22. Information is listed for both insulated and uninsulated ducts. The factors were determined by analyzing data for all combinations of ambient temperature, diameter, insulation, and velocity. The velocity correction factors are, for all practical purposes, independent of ambient temperature and duct diameter. The maximum deviation from the values listed in Table 22 was 3% and the average deviation was 2%. The velocity correction factors for insulated ducts increased slightly with insulation thickness but the maximum deviation from

Table 22

Velocity Correction Factors

Velocity, f.p.m.	400	500	600	700	800	900	1000	1100	1200
Uninsulated Duct	1.60	1.40	1.23	1.10	1.00	0.92	0.85	0.79	0.74
Insulated Duct	1.89	1.57	1.31	1.13	1.00	0.90	0.81	0.74	0.68

the values presented in Table 22 was only 3.5% and the average deviation was 2%. The velocity correction factors permit utilization of the temperature rise data presented in Figure 36 and Tables 20 and 21 for any velocity between 400 and 1,200 f.p.m. To illustrate, assume that in Example 1 the velocity is 500 f.p.m. instead of 800 f.p.m. The velocity correction factor for 500 f.p.m. in uninsulated ducts is 1.40. Therefore, the temperature rise in the first 10-foot segment is 1.40 times 0.19° F. or 0.27° F/ft. The temperature entering the second 10-foot segment would be 55.0° F. plus 10(0.27)° F. or 57.7° F.

4. Ambient Temperature Correction Factors

Ambient temperature correction factors are presented in Table 23. The ambient temperature correction factors are based on 125° F. and 75° F. ambient temperatures. These ambient temperatures correspond to the temperature rise data in Figure 36 and Tables 20 and 21. The ambient temperature correction factors are independent of velocity and diameter. They are dependent upon entering temperature and the amount of duct insulation. Presentation of the data as a function of ambient temperature would require a separate curve for each entering air temperature. However, it was found that the ratio $\left[\frac{\text{ambient temperature} - \text{entering temperature}}{\text{base temperature} - \text{entering temperature}}\right]$ provided an index of the ambient temperature correction factors and left the amount of insulation as the only remaining variable influencing the factors. Therefore, the ambient temperature correction factors are presented as a function of the ratio given above with separate values listed for uninsulated and insulated ducts. The ambient temperature correction factors permit utilization of the temperature rise data presented in Figure 36 and Tables 20 and 21 for any ambient temperature between 75° and 150° F. If the ambient temperature is between 75° and 100° F. the correction factors are based on the 75° F. temperature rise data. If the ambient temperature is between 100° and 150° F. the correction factors are based on the 125° F. data. To illustrate, assume that in Example 1 the ambient temperature is 85° F. instead of 75° F. The correction is based on the 75° F. data. The ratio $\frac{t_A - t_i}{75 - t_i}$ is equal to $\frac{85 - 55}{75 - 55}$ or 1.5. The ambient temperature correction factor from Table 5 is 1.63. Therefore, the temperature rise in the first 10-foot segment of duct is 1.63 times 0.19° F. or 0.31° F/ft.

Table 23

Ambient Temperature Correction Factors

$\frac{T_a - T_i}{125 - T_i}$	100° to 150° F. Ambient Range								
	0.6	0.7	0.8	0.9	1.0	1.1	1.2	1.3	1.4
Uninsulated Duct	0.56	0.66	0.78	0.89	1.00	1.12	1.23	1.35	1.46
Insulated Duct	0.59	0.69	0.79	0.90	1.00	1.10	1.20	1.31	1.41
$\frac{T_a - T_i}{75 - T_i}$	75° to 100° F. Ambient Range								
	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5
Uninsulated Duct	1.63	2.25	2.90	3.50	4.15	4.80	5.45	6.10	6.80
Insulated Duct	1.50	2.00	2.50	3.05	3.55	4.05	4.60	5.10	5.60

D. UTILIZATION OF CURVES AND TABLES

The data presented in Figure 36 and Tables 20, 21, 22, and 23 are sufficient for the determination of the temperature rise occurring in most applications. The solution of a typical problem is presented below to illustrate the combined use of velocity and ambient temperature correction factors.

Example 2. A 6-inch-diameter duct insulated with 1-inch thick glass fiber insulation and covered with a paper vapor barrier (emissivity = 0.90) is located in an attic space where the ambient temperature is 140° F. The duct is 18 feet long. The air temperature at the entrance to the duct is 55° F. and the air velocity is 1,000 f.p.m. Determine the total temperature rise and the exit temperature.

Solution: The velocity correction factor from Table 22 is 0.81. From Table 23 the ambient temperature correction factor for a ratio of $\frac{t_A - t_i}{125 - t_i}$ equal to $\frac{135 - 55}{125 - 55}$ or 1.14 by interpolation is 1.14. From Table 21 for 800 f.p.m. velocity and 125° F. ambient temperature, the temperature rise in the first 10-foot segment is 0.16° F/ft. Applying the corrections factors and multiplying by 10 feet, the temperature rise in the first 10 feet is (0.81) (1.14) (10) (0.16) = 1.47 or 1.5° F. The temperature entering the 8-foot segment is 56.5° F. The velocity correction factor is the same, 0.81. The new ratio must be calculated to obtain the ambient temperature correction factor, $\frac{135 - 56.5}{125 - 56.5}$ or 1.15, and the ambient temperature correction obtained from Table 5 is 1.15 by interpolation. For 800 f.p.m. velocity and a 125° F. ambient temperature, the

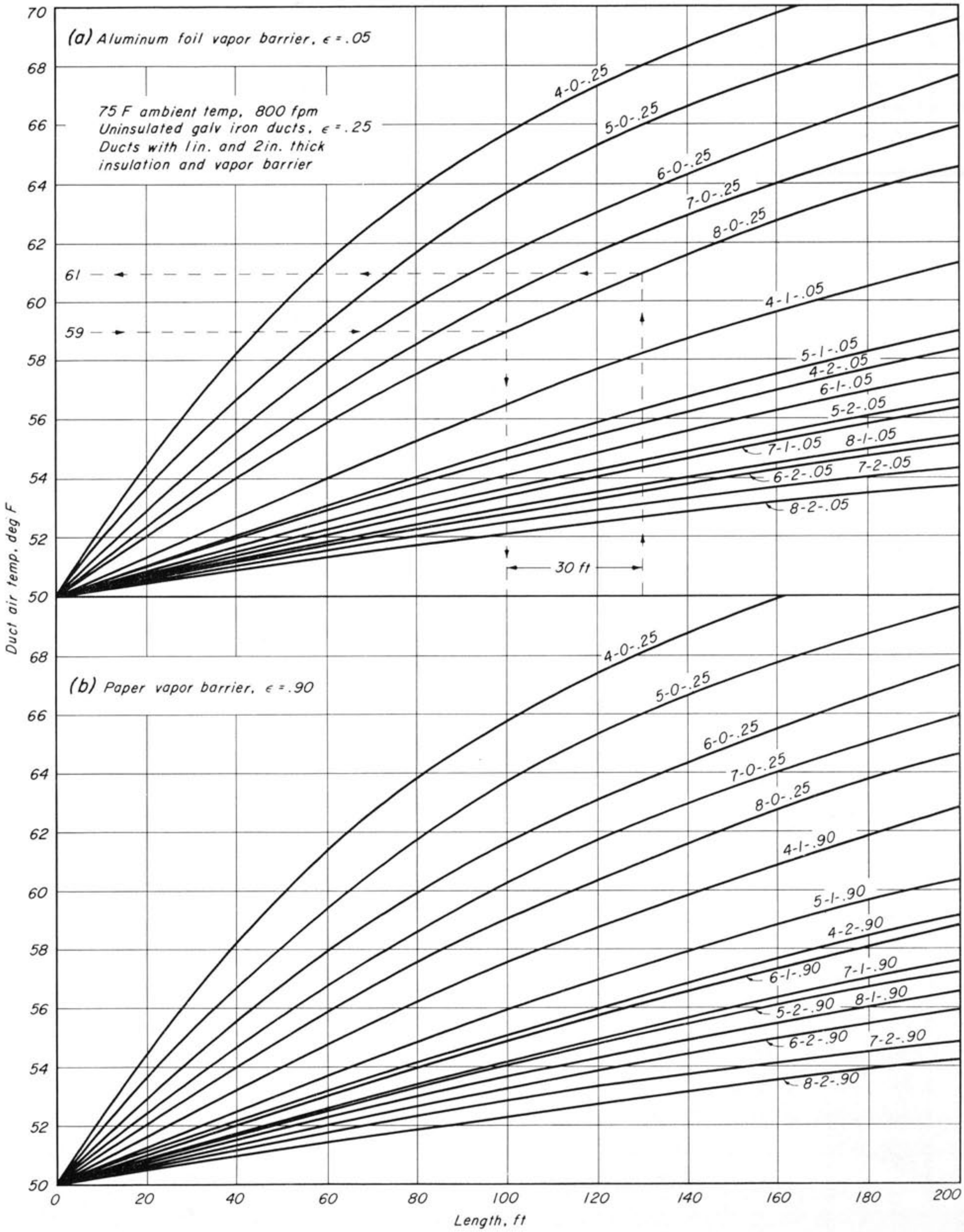


Figure 37. Temperature-length curves for uninsulated ducts and insulated ducts with vapor barriers, 75° F. ambient air temperature

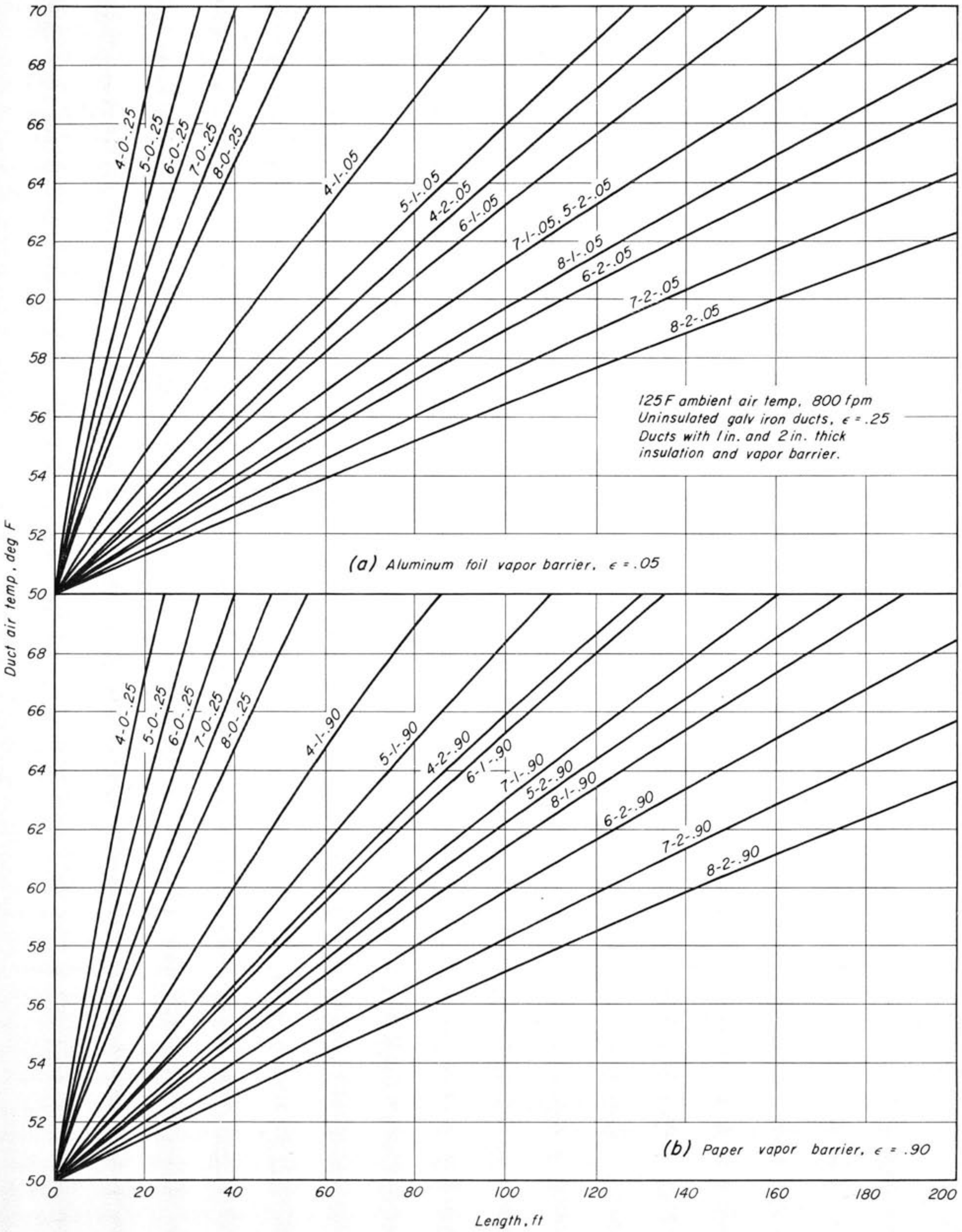


Figure 38. Temperature-length curves for uninsulated ducts and insulated ducts covered with aluminum foil vapor barriers, 125° F. ambient air temperature

rise in the last segment is 0.15° F/ft. Applying the correction factors and multiplying by 8 feet, the temperature rise in the last 8 feet of the duct is (0.81) (1.15) (8) (0.15) or 1.1° F. The exit temperature is 56.5° F. plus 1.1° F. or 57.6° F. and the total temperature rise is 2.6° F.

E. TEMPERATURE-LENGTH CURVES

Determination of the temperature rise in ducts is more convenient when the data are presented in the form of temperature-length curves as shown in Figure 37 and Figure 38. These curves were developed from the temperature rise data presented in Figure 36 and Tables 20 and 21 and are valid for 800 f.p.m. velocity and either 75° or 125° F. ambient temperature. Figures 37a and 37b are for 75° F. ambient temperature and Figures 38a and 38b are for 125° F. ambient temperature. Figures 37a and 38a are for uninsulated ducts and insulated ducts with aluminum foil vapor barrier ($\epsilon = 0.05$). Figures 32b and 33b repeat the data for uninsulated ducts and contain data for insulated ducts with paper vapor barrier ($\epsilon = 0.90$). The duct diameter, insulation thickness, and exterior surface emissivity are indicated for each curve. For example, 8-1-.05 indicates an 8-inch diameter duct with 1-inch thick insulation and an aluminum foil vapor barrier (emissivity = 0.05). The emissivity of a paper vapor barrier is 0.90 and the surface emissivity of a galvanized iron duct is 0.25. The following example, illustrated in Figure 37a, shows how the temperature-length curves may be utilized.

Example 3. Assume that 59° F. air enters an 8-inch-diameter galvanized iron duct which is uninsulated and 30 feet in length. The ambient air temperature is 75° F. Determine the temperature of the air leaving the duct. The arrows on Figure 37a indicate the use of the curves to determine the final temperature. Enter the chart on the ordinate at 59° F. (entering temperature). Proceed horizontally to the duct specification curve (8-0-.25 curve for this example). From the intersection with the curve proceed downward to the length scale on the abscissa (100 feet in this case). Add the length of the duct to the intersected length (100 + 30 = 130 feet). Proceed upward on the 130-foot length line and again intersect the 8-0-.25 curve. From this intersection proceed to the left to the ordinate and read the leaving temperature (61° F. in this case).

Ambient temperature and velocity corrections factors are not strictly applicable to temperature-length curves but if the ducts are less than 50 feet long the error introduced will be small, less than 5% for insulated ducts and less than 20% for uninsulated ducts. Larger errors occur for smaller ducts and lower velocities. Specific examples of the variation between the temperature rise data obtained by temperature-length curves for other than 800 f.p.m. velocity and 75° and 125° F. ambient temperatures are included in the following section.

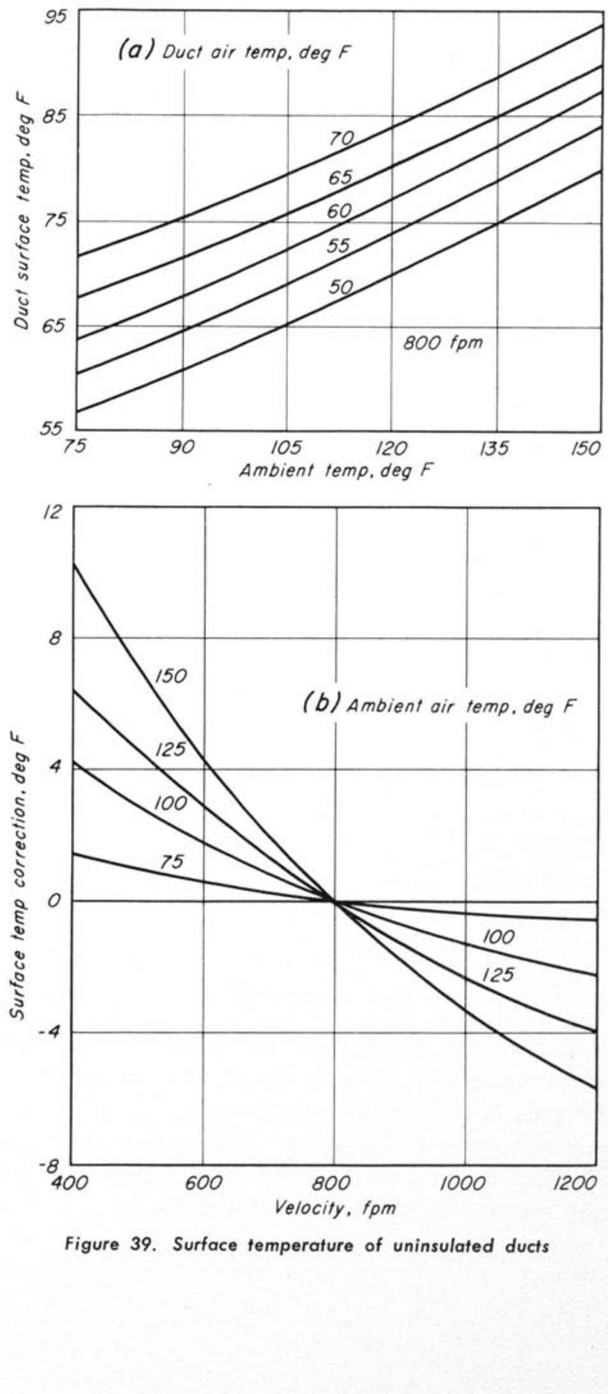


Figure 39. Surface temperature of uninsulated ducts

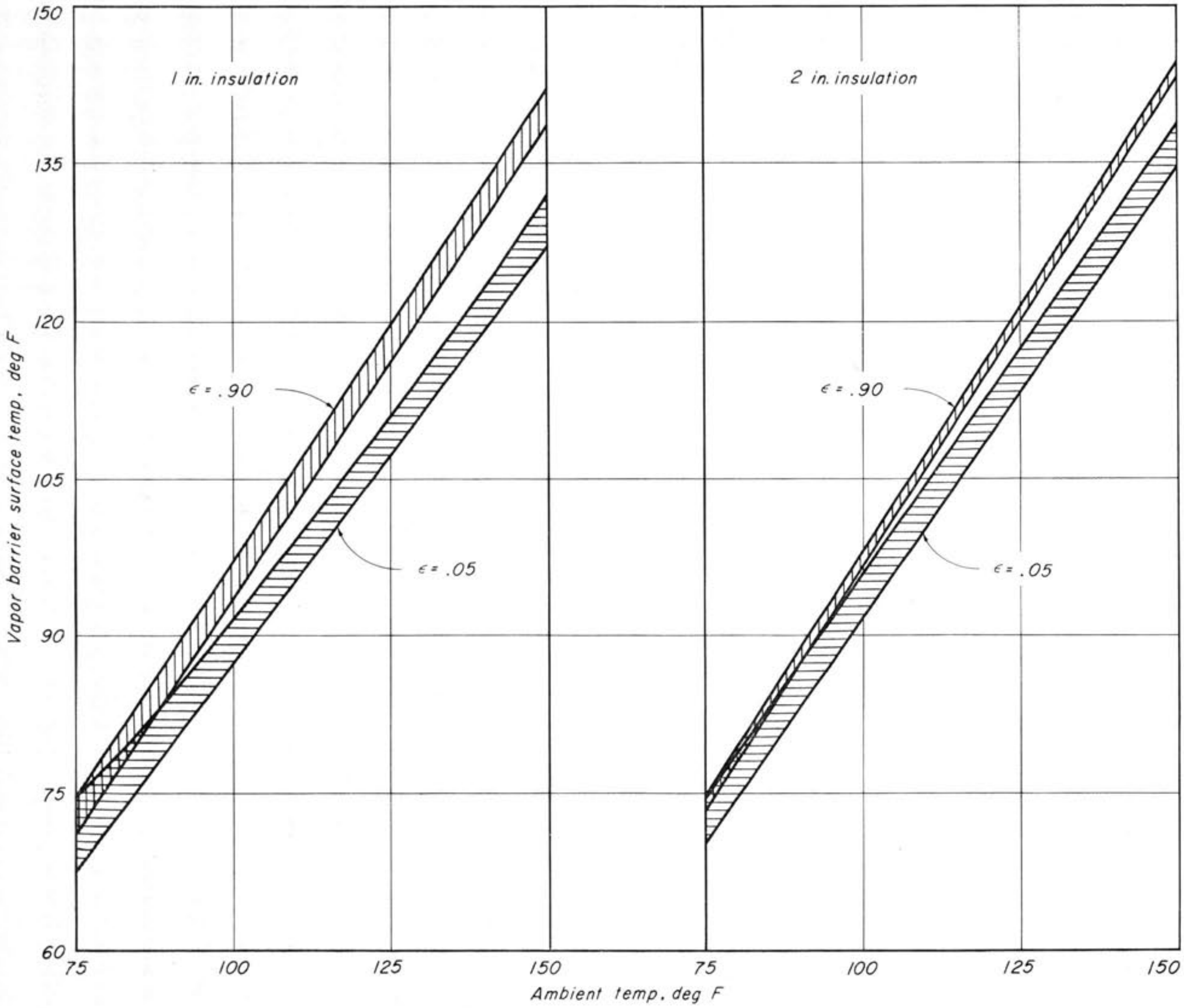


Figure 40. Vapor barrier surface temperature of insulated ducts

Adjustments to other than the base ambient temperatures and velocity are made in the following manner: Obtain the temperature rise from the temperature-length curve for the given duct length as described above. Determine the velocity correction factor from Table 22. Calculate the ratio $\frac{t_A - t_i}{\text{base temp} - t_i}$ and obtain the ambient temperature correction factor from Table 23. Multiply the temperature rise obtained from the temperature-distance curve by the velocity and ambient temperature correction factors. The temperature rise obtained by this method will be sufficiently accurate for all but the most precise designs.

F. CONDENSATION ON DUCT SURFACES

The data presented in the preceding sections are based upon the assumption that condensation does not occur on the duct or insulation surfaces. This will be true if the exposed surface temperature is above the dew point of the ambient air. Figures 39 and 40 show the surface temperatures (duct surface temperature for uninsulated ducts, vapor barrier surface temperature for insulated ducts). The surface temperatures are independent of duct diameter.

For uninsulated ducts, the surface temperature is a function of the duct air temperature, ambient temperature, and duct air velocity. The surface temperatures of uninsulated ducts exposed to ambient temperature from 75° to 150° F. are shown in the upper part of Figure 39 for a duct air velocity of 800 f.p.m. with duct air temperature as a parameter. The surface temperatures range from 56° to 94° F. For duct air velocities other than 800 f.p.m. the surface temperature must be adjusted by the correction indicated in the lower part of Figure 39. The surface temperature correction varies from 10.6° to −5.7° F. depending upon the velocity and ambient temperature.

The vapor barrier surface temperatures for insulated ducts are shown in Figure 40. The surface temperature was practically independent of duct air temperature and velocity within the range investigated. It is principally dependent upon ambient temperature, surface emissivity, and insulation thickness. The surface temperatures are shown as a function of ambient temperature with bands for each vapor barrier emissivity encompassing all variations due to duct air temperature and velocity. The lower emissivity, in conjunction with reducing the temperature rise in ducts, causes a lower surface temperature. The lower part of the bands are for lower duct air temperature and higher velocities. As stated earlier, condensation will occur if the exposed surface temperature is equal to or less than the dew point of the ambient air.

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